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MACHINE DESIGN

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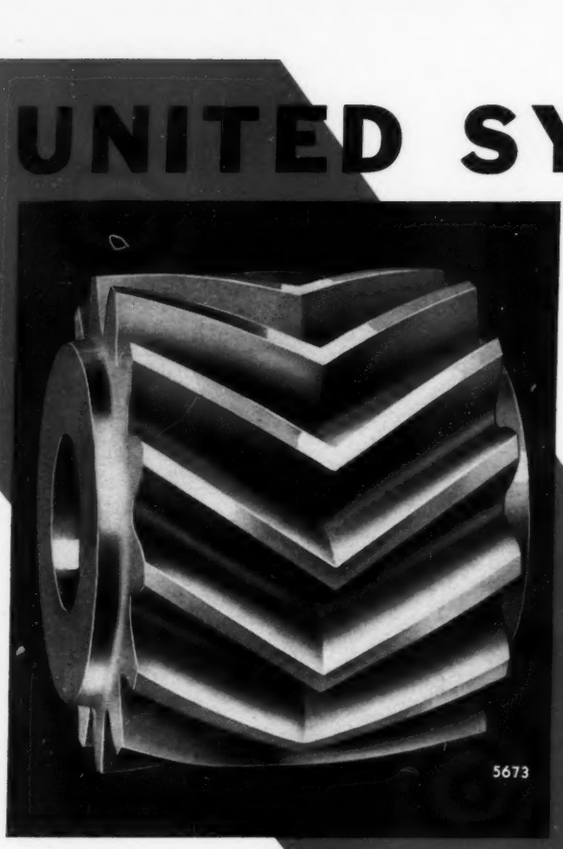
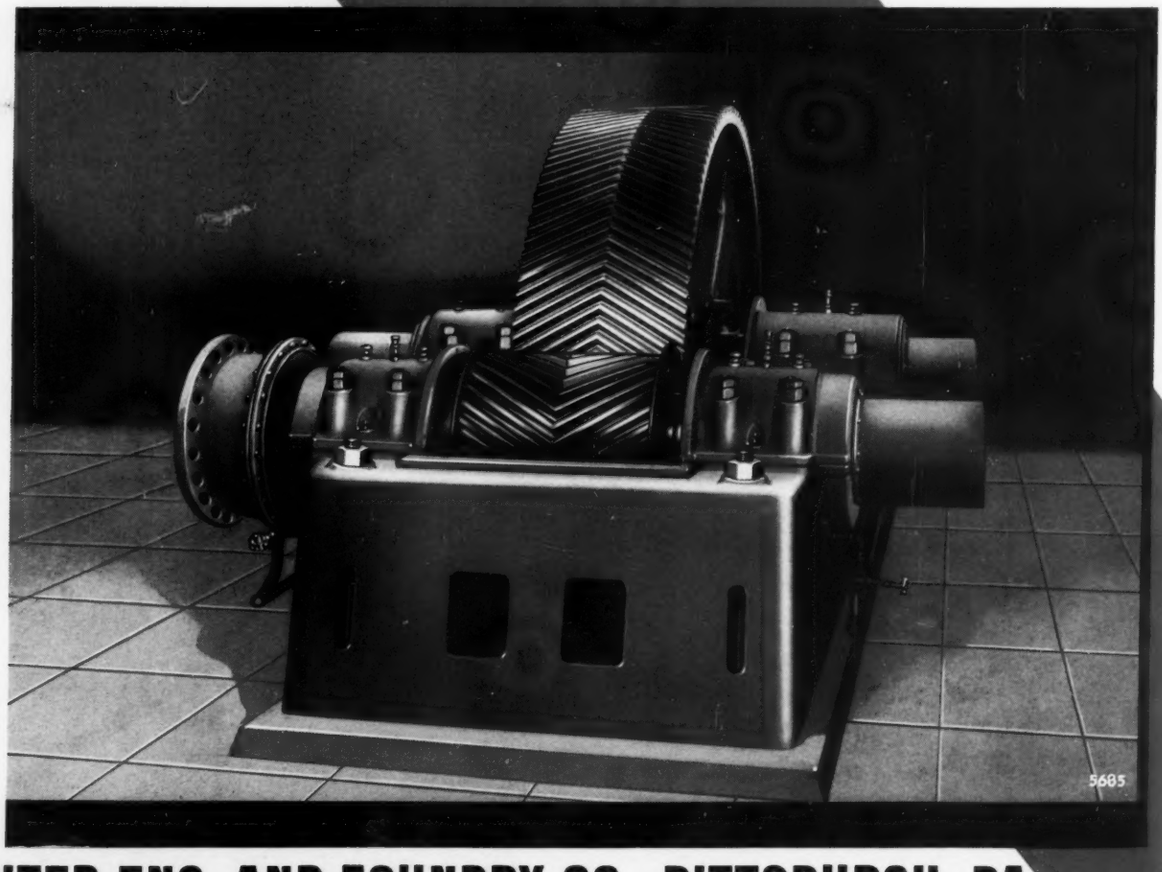
UNITED SYKES GEARS



UNITED Sykes Gears from 3" to 22 ft. diameter and with load capacity of from 50 to 10,000 H.P. are fully generated herringbone gears of the most modern type. The tooth contours are within .001" to .003" of accuracy even on gears of 6" C.P. and the helical trace of the teeth is the most accurate produced by any gear machine. The teeth are the strongest made, wide at the base, with generous fillets and are unbroken across the entire face. They are applied in a series of standard and special gear frames of either iron cast or welded steel and can be depended upon to operate continuously for an indefinite period without appreciable wear or fear of breakdown. Pinions made from UNITED special alloy surface hardening steel have no distortion, have a surface hardness of approximately 475 Brinnell, and a depth of case of nearly 1/8".

50 H.P.— TO —10000 H.P.



UNITED ENG. AND FOUNDRY CO., PITTSBURGH, PA.

MACHINE DESIGN

FRANKLIN H. JOHNSON
Publisher

VOLUME VI JUNE, 1934 NUMBER 6

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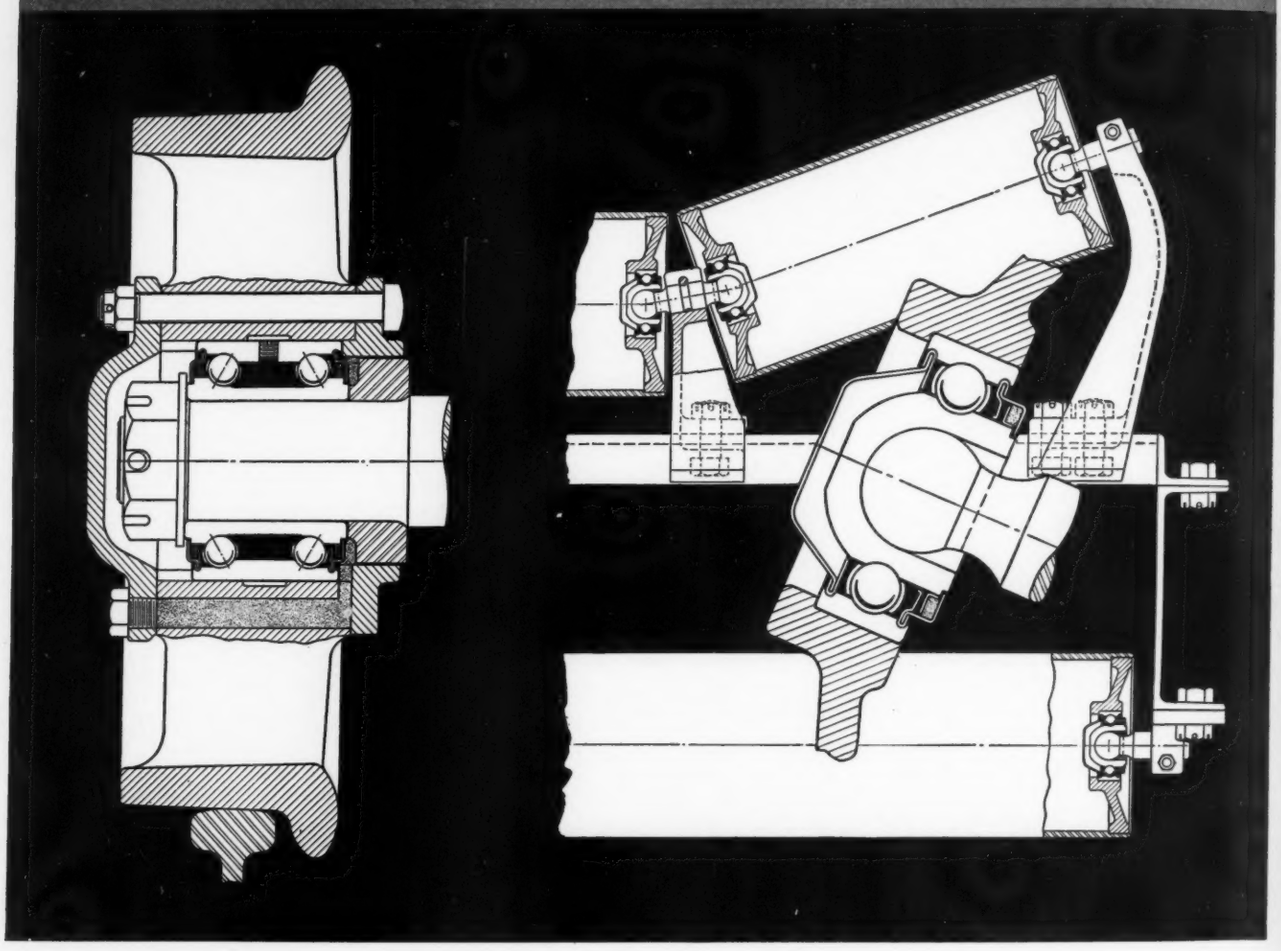
Self-enclosed Bearings
simplify and improve design



A HIT AT THE MINE SHOW

OF outstanding interest at the recent show in Cincinnati were these two new self-sealed, self-lubricated ball bearings for mine car wheels and conveyor rolls. » » » They initiate a trend which is bound to change the design of all future mechanism of this type. The interesting details are given in booklets entitled "Lubricated for Life", which will be mailed upon request. » » » The New Departure Manufacturing Company, Bristol, Connecticut. Branches at Detroit, Chicago, San Francisco.

MINE CAR BEARINGS

An extra wide double row ball bearing permanently sealed on both sides and lubricated for life. As a further guard against flooding with water, provision is made for an additional exterior seal of heavy grease.

CONVEYOR BEARINGS

Completely self-sealed and lubricated for life. Most economical bearing for this work . . . positively excludes all injurious substances . . . requires 60 to 80% less parts in average conveyor.

NEW DEPARTURE

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Classified for Convenience when Studying Specific Design Problems

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The "Izod" test

ELIMINATES TEMPER EMBRITTLEMENT IN STEEL

Embrittlement is the bugaboo of high-temperature steels, and the cause of much trouble when drawing other constructional steels. It can be conquered! A small amount of Molybdenum will retard temper embrittlement, while a .75% addition will completely eliminate this dangerous trouble.

The following comparison of two similar steels, with and without "Moly," shows the effects of this extraordinary alloy when one sample is cooled slowly and another water-quenched after tempering at 1150° for two hours:

Analysis			*Impact (foot pounds)*	
C	*Mn*	*Mo*	*Cooled in water*	*Cooled slowly*
.36	2.24	Nil	74	4
.30	2.30	.52	59	50

In high-temperature installations, a similar condition is often encountered. Why take a chance when the difficulty is so easily and economically overcome?

Molybdenum is one of the greatest discoveries in alloy history. Its many-sided qualities for improving steel and iron have excited the wonder of the engineering and metallurgical world. Resistance to fatigue, shock, creep, abrasion, corrosion and temper embrittlement has definitely been increased under Moly formulae repeatedly tested and proved in service. Machinability of heat-treated steels is economically increased without loss of strength or other essential qualities. *Write for our latest book on Molybdenum.* Discuss your alloy problems with our modern experimental laboratory and engineering service. Climax Molybdenum Company, 295 Madison Ave., New York.

CLIMAX Mo-lyb-den-um

This PROVED SPEED CONTROL

provides ANY SPEED desired—and maintains it WITHOUT FLUCTUATION

New REEVES Variable Speed Transmission (enclosed design, *cover removed)* with "centralized" force-feed lubrication, new Endless Cord V-belt and new "drop-in" frame construction, permitting easy removal of operating parts.

A variable speed control unit must do MORE than provide an infinite number of operating speeds for the machine it regulates. It must ACCURATELY MAINTAIN any speed setting as long as desired—*regardless* of how fast or slow the speed—and *regardless* of varying loads.

The REEVES Variable Speed Transmission *does* this. The REEVES *absolutely* holds any desired speed with hair-line accuracy, due to positive delivery of power by the *time-tested* REEVES V-belt whose tension is automatically controlled through the patented REEVES equalizing lever link.

Many other fundamental and exclusive advantages in design, construction and performance explain the world-wide popularity of the REEVES Transmission. Over 80,000 applications in 16,000 industrial plants. Used as standard equipment by 900 different machine builders. For every need in machine design, there is a REEVES in the *correct* size, speed ratio, control and design, including an internal-operating-parts mechanism for incorporation as an integral part of a machine.

SEND FOR THIS BOOK

REEVES PULLEY COMPANY, Columbus, Ind.

Send, without obligation, information on designing variable speed control into machines, as contained in your latest Catalog H-99.

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Company.....

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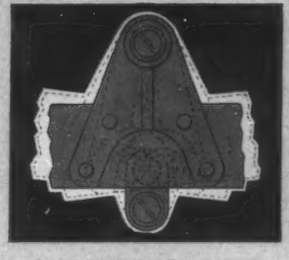
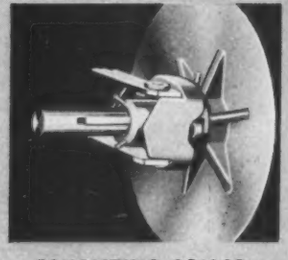
A FEW OF MANY ★ REEVES ★ ENGINEERING FEATURES



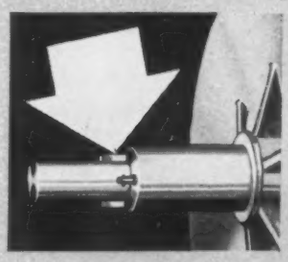
EQUALIZING LEVER LINK

Automatically maintains uniform V-belt tension regardless of diameters of arcs of contact assumed by belt. No speed fluctuation under varying loads such as results by allowing belt to take different arcs of contact under spring tension.



EQUALIZING COLLAR

Provides four-point universal mounting on each ball thrust bearing. Distributes load evenly on each ball in each bearing; no cocked links nor multiplicity of joints. Holds bearing in perfect alignment with discs at all speed settings.



SHAFT DRIVING KEYS

Two large driving keys in each shaft. For 50 H.P. units and larger, four keys in each shaft. Keys are securely imbedded, accurately aligned and fitted to keyways in discs hubs, permitting free lateral movement of discs on shafts.

COMPACT MOTOR MOUNTING

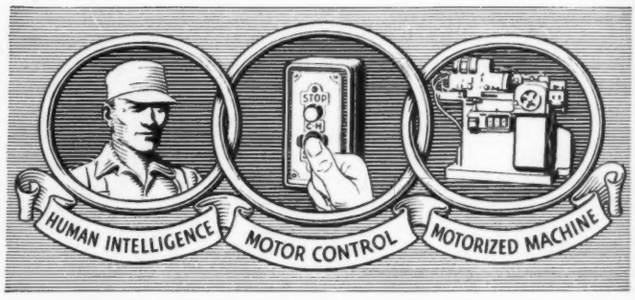
Universally adjustable motor base obtainable for any REEVES unit. When compared with any other speed varying equipment—electrical or mechanical—on a capacity and safety factor basis, REEVES is the most compact.

LINKING TRAINS brings *SPEED* with *SAFETY*

MOVING units, whether they are railroad trains or machines in Industry must be linked to human intelligence. Modern railroading depends on communication for fast, accurate control. In Industry the *vital* link is Motor Control.

Being the *vital* link in the mastery of man over machines, Motor Control is far more important than either its size or cost indicates. It is a factor in the speed and dependability of production, the quality of workmanship, the cost of power and maintenance, the safety of men, motors and machines.

Wise builders of motor-driven machines never lose sight of this important fact. They see Motor Control as much more than a small, low-cost unit in their designs. They recognize Motor Control as the one link between human intelligence and machine performance. They refuse to hazard the consequences of *indifference* toward Motor Control.



The designing and building of Motor Control is a separate and distinct division of electrical engineering. The place to go for consultation and advice on Motor Control problems is to a control specialist. Cutler-Hammer indisputably qualifies as such a specialist. Your problems will be welcomed here. A letterhead request brings the C-H Catalog.

Standardize on Cutler-Hammer Motor Control to insure the performance of your machines and to add another sales feature at no expense. You, too, should call in C-H engineers while designs are in the blueprint stage. CUTLER-HAMMER, Inc., *Pioneer Manufacturers of Electric Control Apparatus,* 1310 St. Paul Ave., Milwaukee, Wis.

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C-H

The Control Equipment Good Electric Motors Deserve

(B-219)

SIMPLE ARITHMETIC

Yet Vital to Spring Performance

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1831

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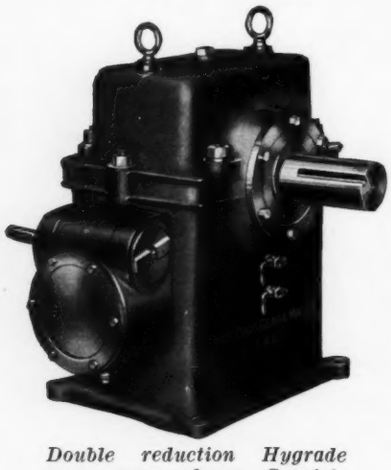
There's an IXL UNIT

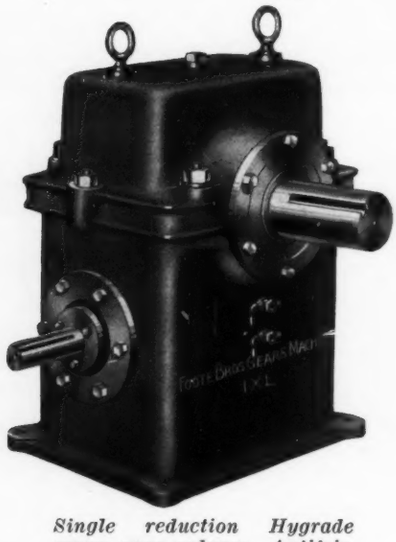
Type HGM radiating worm powered gear. Motor is supported from the reducer. Exclusive radiating design provides heat radiating corrugations. Output shaft speeds range from 14.4 R.P.M. to 600 R.P.M. Power capacities from ½ H.P. to 30 H.P.

Double and triple reduction helical powered gears. Concentric shaft gives straight-through drive. Motor is mounted on reducer frame. Output shaft speeds from 6.08 R.P.M. to 170 R.P.M. Power capacities from ¾ H.P. to 75 H.P.

IXL spur reducers. Compact, fully enclosed, self-lubricating, non-planetary reductions with high and low speed shaft in direct line. Ratios 4 to 1 up to 350 to 1 in sizes up to 100 H.P.

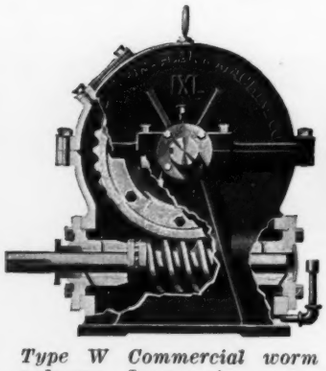


Double reduction Hygrade worm gear reducer. Consists of two worm and worm gear reductions assembled as one unit. These units are standard and can be furnished in any size with practically any ratio desired.



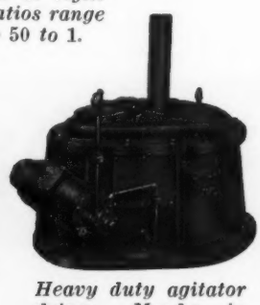
Single reduction Hygrade worm gear reducer. Antifriction bearings throughout. Most economical and efficient unit for transmitting power and reducing speed at right angles. Standard ratios range from 4 to 1 up to 50 to 1.

Single reduction Hygrade vertical worm gear reducer. Ideal unit for agitators and mixers. Worm is horizontal and worm gear shaft is vertical. Ratios range from 4 to 1 up to 97 to 1.

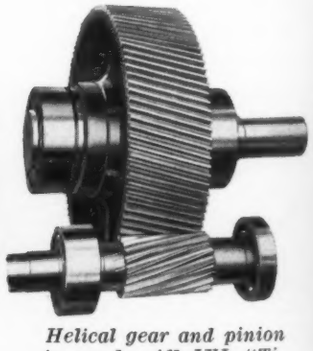


Type W Commercial worm reducers. Inexpensive means of speed reduction. Compact, fully enclosed and self-lubricating. Ratios range from 7 to 1 up to 120 to 1. Capacities from fractional H.P. to 150 H.P.

Industrial heavy duty helical speed reducers. For oil well pumping and industrial applications where heavy shock and load reverses require rugged design and heavy bearing construction.



Heavy duty agitator drives. Made in double and triple reductions. Ruggedly built to support weight of agitators. Step down speeds of 1800 or 1200 R.P.M. to 5 R.P.M. to 35 R.P.M. Capacity range from 5 H.P. to 25 H.P.



Helical gear and pinion in mesh. All IXL "Titan" Helical speed reducers are designed to give maximum operating efficiency and durability. Proper design results in lighter more compact and more efficient units.

for Every Application!

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DESIGNING engineers in many different industries depend on Foote Bros. for their speed reduction requirements. They know that Foote Bros. make IXL speed reducers in hundreds of types and sizes to fit the particular need of the designer. Foote Bros. recommendations are unbiased and unprejudiced, because somewhere in the IXL line is exactly the right unit for any particular purpose. The Foote Bros. book "Gear Problems" is a 664-page designing engineer's text book. Every man who is designing equipment should have a copy of this book on his desk. As useful as your slide rule. The coupon below —

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Seven Reasons Why Helical Gears Are Preferable for Speed Reducers and Powered Gears

1. Greater accuracy because generated by hobbing process.
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7. Gears can be heat-treated to greater hardness before cutting or heat-treated after cutting with less distortion.

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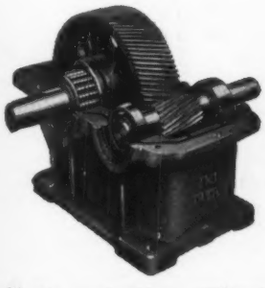
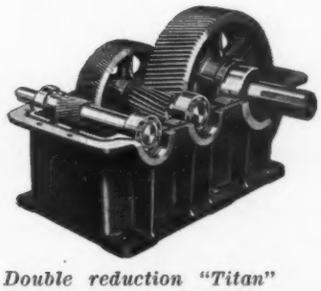


Single reduction "Titan" Helical gear speed reducer. Smooth, continuous, noiseless action. All "Titan" Helical gears are heat treated and mounted with anti-friction bearings



Double reduction "Titan" Helical gear speed reducer. All horse power ratings of "Titan" Helical units are 25% greater than the herringbone units, size for size.

Use this Coupon for new 664-page "Gear Problems"

(For Executives and Engineers)

Foote Bros. Gear & Machine Co.
5303 S. Western Blvd.
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Please send me, without obligation, copy of the 664-page designing engineer's text book, "Gear Problems."

Name .....

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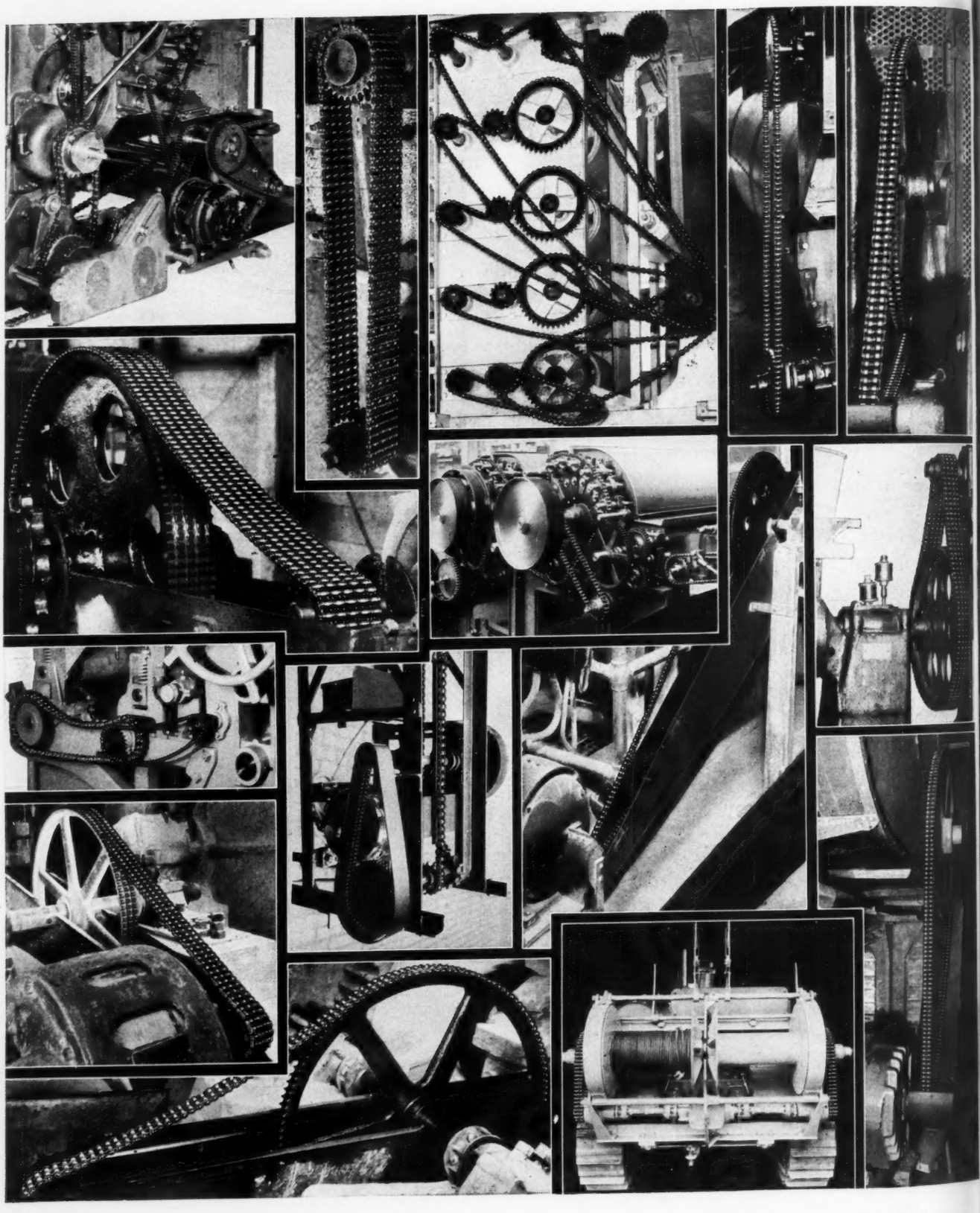
DIAMOND CHAIN



..Most Universal of Drives

Diamond Roller Chain Drives are efficient and economical for practically every power transmission problem in industry—making Diamond Chain the most universal of drives. The illustrations shown indicate the tremendous diversity of application.

There are thousands of Diamond Drives operating at speeds up to 3600 R.P.M.—a considerable number at even higher speeds. Crawling speeds and intermittent operation are also efficiently handled with Diamond from fractional horsepower capacities up.

Center distance is no obstacle with Diamond Chain, for chain drives are positive—continuously. Speed ratios do not vary, and there is practically no let-down of efficiency.

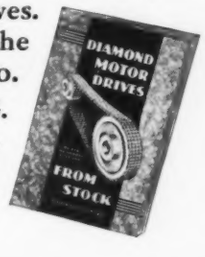
Diamond Chains drive on either side, running either over or under sprockets so that relative position of machine units is no problem. A unit can be placed where logic rather than limitations of the drive dictates.

Greater Horsepower per Unit Width

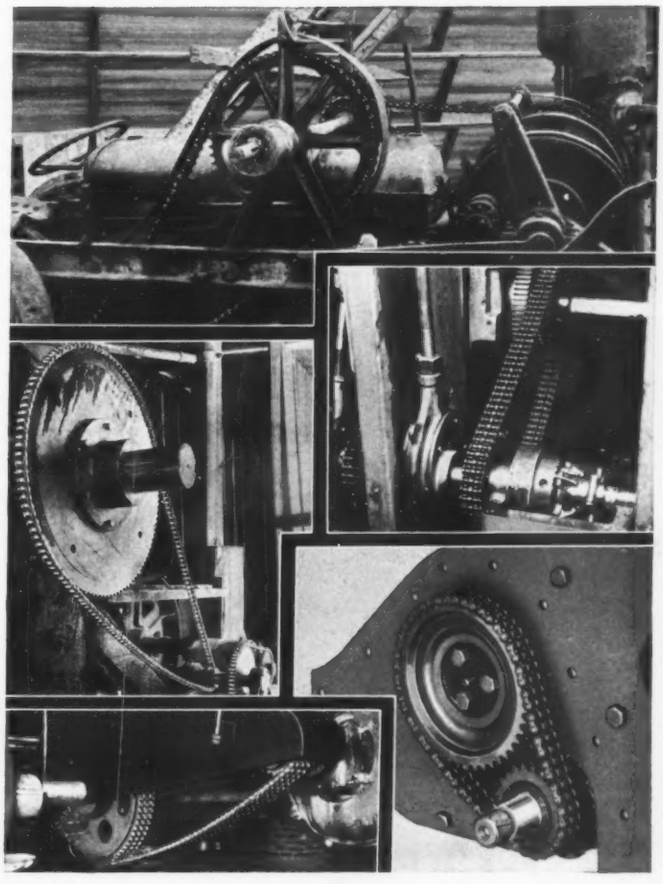
They require much *less space per hp. transmitted,* are simple to install—and easy to maintain.

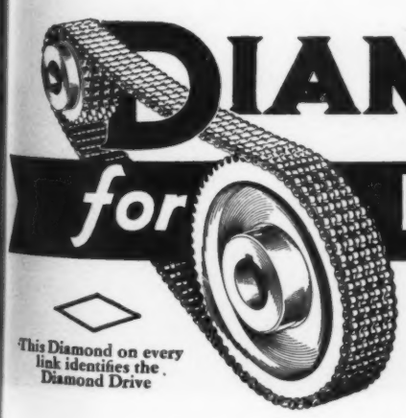
Diamond Chain Drives are carried in stock complete by Diamond Distributors located in all principal cities.

You can reduce power losses and simplify your drive problems with Diamond Drives. Start by mailing the coupon for the Diamond Stock Drive Catalog No. 78. DIAMOND CHAIN & MFG. CO., 435 Kentucky Ave., Indianapolis, Ind. *Offices & Distributors in All Principal Cities.*



High Speed and Low Short Centers and Long Up to 672 H. P.





This Diamond on every link identifies the Diamond Drive

DIAMOND ◇ DRIVES

for Every Industrial Need

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Gentlemen: Please mail a copy of your Diamond Stock Drive Catalog No. 78.

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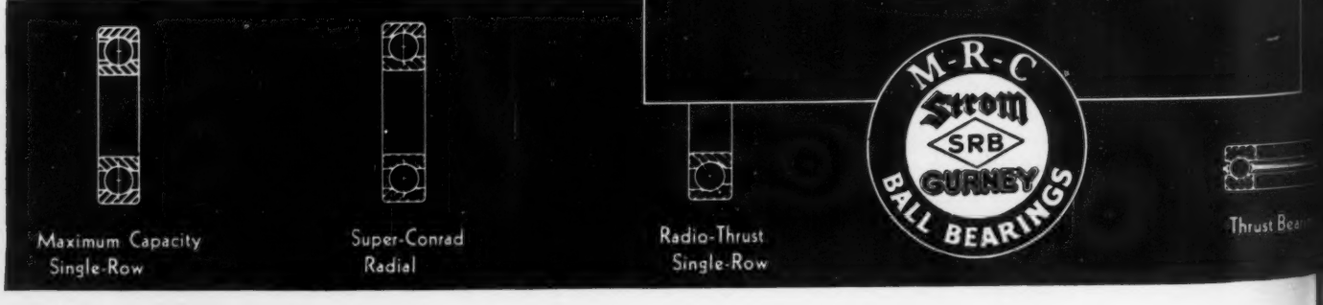
M-R-C Leadership

. . . YOU

One of the Leaders—Typical of the complete M-R-C Line

M-R-C DUPLEX BALL BEARING—GURNEY TYPE

This bearing consists of a perfectly matched pair of Radio-Thrust Bearings. It will carry exceptionally heavy radial loads in combination with thrust loads in either direction. Due to its preloaded rigidity and subsequent freedom from all endwise movement, the Duplex is recommended for machine tool spindles, and in positions where it must sustain severe reversing thrust loads.



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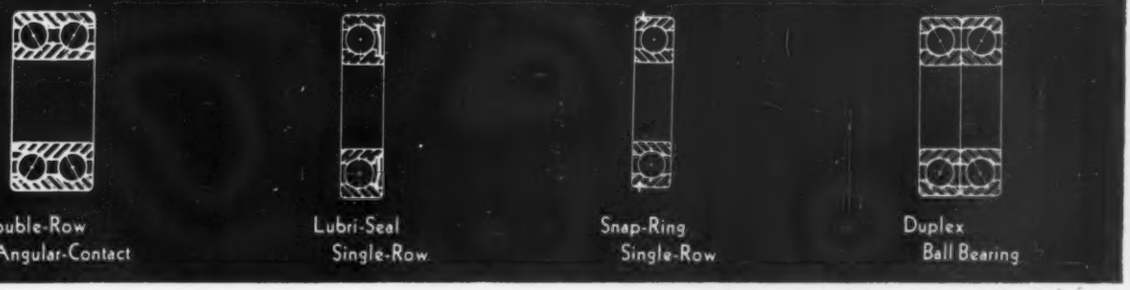
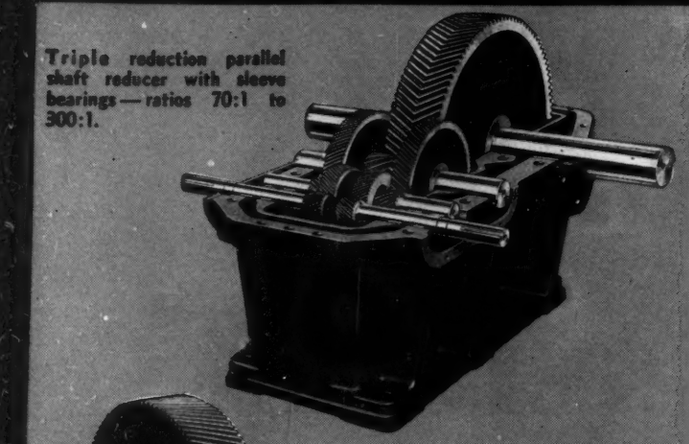
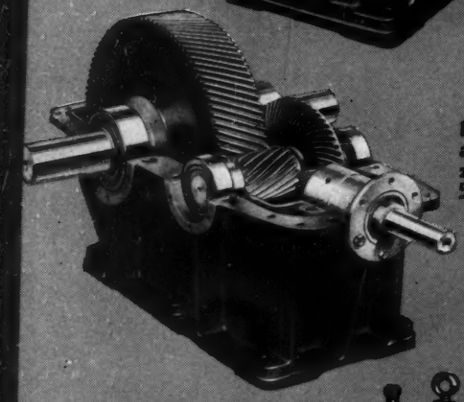
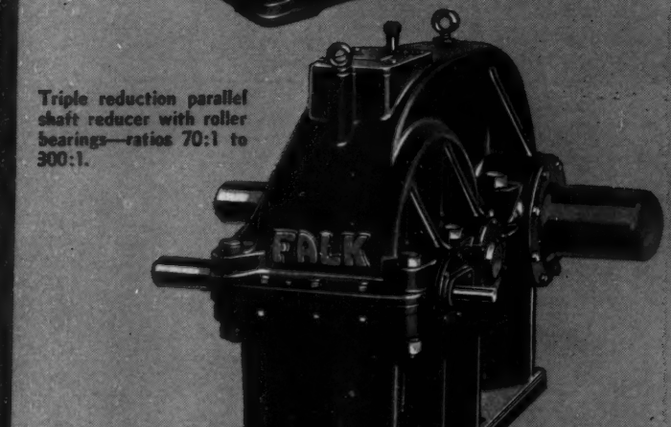




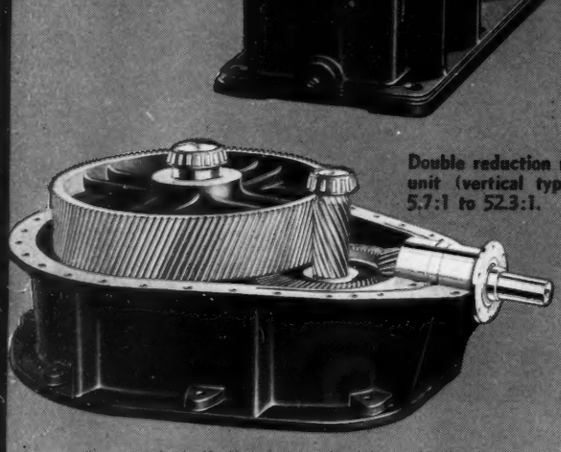
Triple reduction parallel shaft reducer with sleeve bearings — ratios 70:1 to 300:1.



Double reduction right angle reducer (horizontal type)—ratios 5.7:1 to 52.3:1.



Triple reduction parallel shaft reducer with roller bearings—ratios 70:1 to 300:1.



Double reduction right angle unit (vertical type)—ratios 5.7:1 to 52.3:1.

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●

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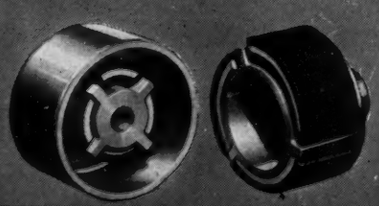

Cuteway view of Falk Flexible Coupling, which combines Flexibility with Torsional Resiliency.

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Write for Bulletin Number 270.

VERTICAL UNITS

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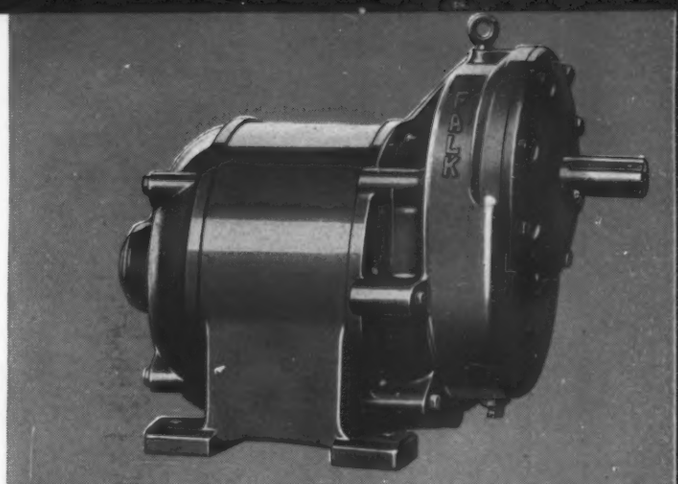
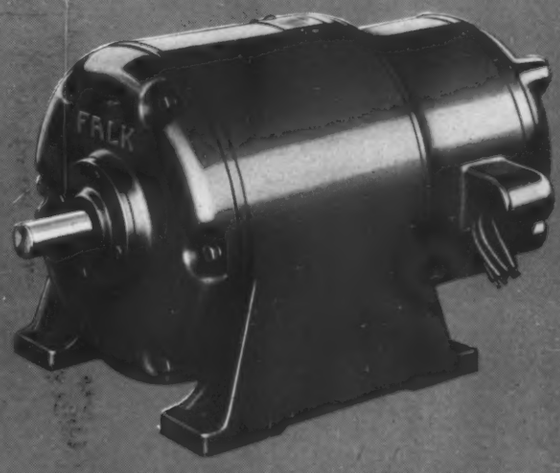
This line includes Geared-Head, Integral and All-Motor type "Z" . . . the latter using any standard horizontal ball bearing type motor with feet and having resilient connection through the Falk-Bibby coupling to the gearing.

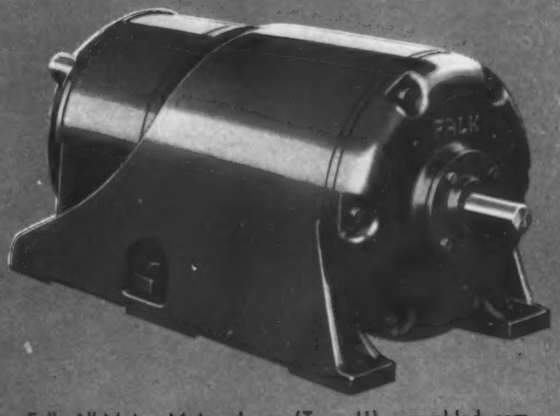
Write for Bulletin Number 271.



Falk Geared-Head Motoreducer (Type L) assembled complete with motor. Ratios 1.5:1 to 9.0:1.



Falk Integral Motoreducer (Type I) assembled complete with motor. Ratios 4.6:1 to 288:1.



Falk All-Motor Motoreducer (Type U) assembled complete—viewed from gear end. Ratios 4.6:1 to 288:1.

IT PAYS TO CONSULT FALK FIRS

Falk Vertical Motoreducers —Integral and All-Motor Types. Ratios 4.6:1 to 288:1.

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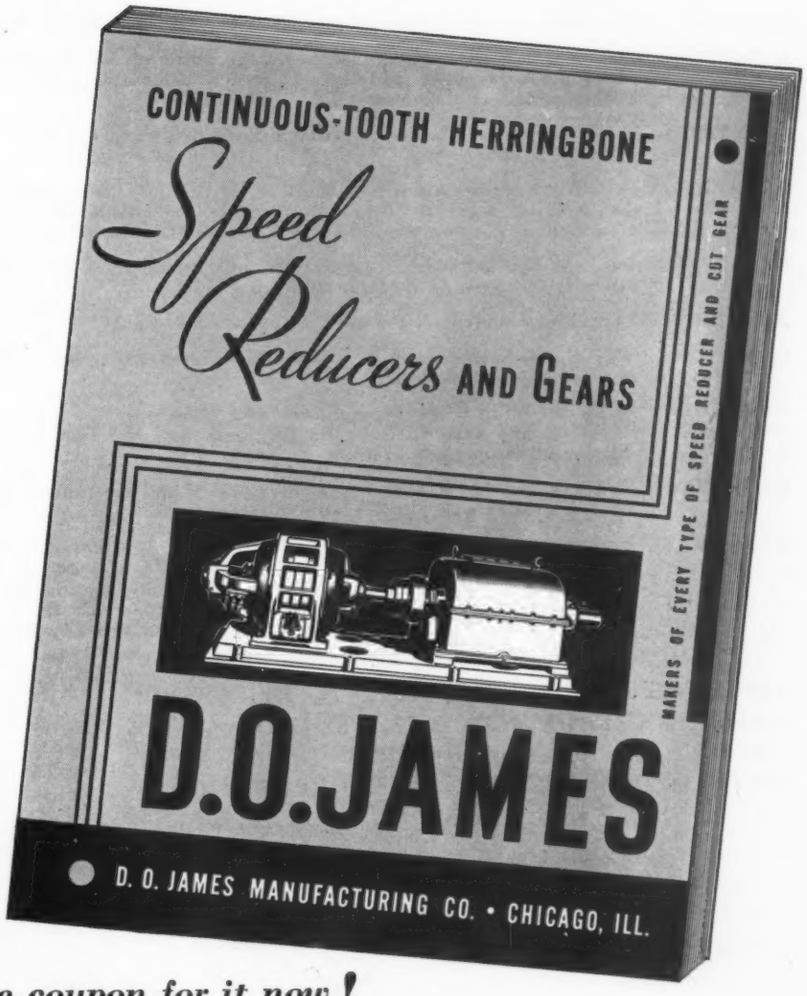
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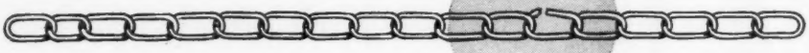

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ANSWER: In G-E All-rubber Cord Sets there is no point of attachment . . . they are not made in several parts ready for attachment, but are of one-piece construction, *The plug is moulded on the cord.*

QUESTION: How do G-E All-rubber Cord Sets and moulded rubber devices differ from those of other makes?

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QUESTION: How can one best avoid appliance failures and consumers' ill-will for his products in the selection of accessories?

ANSWER: Investigate the line of G-E All-rubber Devices — *moulded on the cord.* All types of rubber plugs, strain reliefs, connectors, angle caps, and Delta connections are available . . . and best of all, they are all *moulded on the cord.* They are approved by the Underwriters and not only meet but surpass all tests and requirements.

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THE JOHNSON PUBLISHING CO., CLEVELAND, OHIO
June, 1934 Vol. 6—No. 6

Solving Corrosion Problems with Ceramic Materials

By J. M. W. Chamberlain
Manager Engineering Department, United States Stoneware Co.

STONEWARE, glass and porcelain parts are all found as constituents of various machines, particularly those used in the chemical industry. However, the design technique necessary to embody such ceramic parts in machines is not well understood by many engineers as equipment of this type has been produced by comparatively few companies. Some of the properties of ceramic materials if more generally comprehended would permit of their broader use in the fabrication of those parts that must withstand corrosive influences. Ceramic materials as a class have a more universal resistance to corrosion and erosion than any other known substance.

Speaking in a general way we may say that the ceramics, stoneware, porcelain and glass are immune to the destructive effects of corrosion, but in comparison to metallic materials they are fragile. To help in the understanding of this fragility and to show the methods of minimizing it are the purposes of this article.

When one examines the physical properties of any of the good grades of stoneware, porcelain or glass, he finds that there is but one weakness, all other factors being equal to or better than those of many of the commonly used materials of machine construction. The one weak property which accounts for our fundamental notion that ceramics are frail materials is the fact that they are all cursed with very high elastic moduli. From a structural point of view the problem resolves itself into one of designing around an undesirable elasticity. From the fabrication angle we must know the means of accurately shaping the burned parts in spite of another property, great hardness.

The fact that ceramics are extremely hard is, as a rule, a blessing rather than a curse once the part has been produced. It perhaps would be well to note that it is quite possible to reduce both the modulus of elasticity and the hardness of clay ceramics. To alter them by the known methods of present

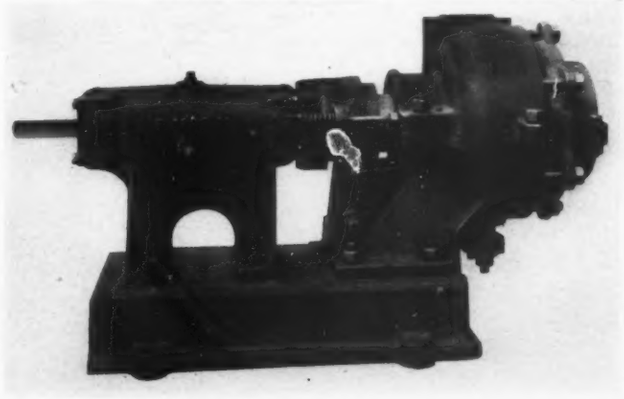

Fig 1—Acid pump employs advantages of ceramic material while minimizing negative qualities

day technique however, renders them of little or no value as parts for machines as their strength is greatly impaired by so doing.

The range of physical properties that it is possible to obtain with chemical stoneware is indicated in Table I. Data is shown only for chemical stoneware since it is possible to vary its properties over a wider usable range than is the case with other ceramic materials.

The hardness of ceramic materials makes it mandatory that they be ground if it is necessary to remove stock or alter dimensions after they have been fired. This grinding can be accomplished by the use of proper carborundum wheels. In order to minimize the amount of grinding it is possible to turn cylindrical parts of small size down to approximate dimensions before they are burned. This is not possible on large parts as the warpage encountered on firing them would not permit of finishing to set dimensions.

High Percentage Contact Necessary

In order to take full advantage of the relatively high compressive strength of chemical ware it is necessary that load bearing surfaces have a high percentage contact. Uneven surfaces or load bearing surfaces with foreign material between them will cause premature failure. This is due to the high elastic modulus which does not permit normal load distribution over the contact face. The high unit loading stress which follows from rigid point contact caused by rough surfaces will, of course, result in an apparently low overall strength. This trouble is corrected by grinding load bearing surfaces and placing gaskets between them. Failure to reckon with this factor causes most of the mechanical failures encountered with clay products.

In original designs involving clay parts it is apparent from the physical data given that the material should be kept in compression loading so far as is practical.

Safety Factors Should Be High

Safety factors, based on average physical data obtained from tests on a given ceramic material, should be higher than those used for metallic materials. This is due to the unfortunate fact that it is not possible to obtain the uniformity in clay materials to which one is accustomed in the metals. The scattering of individual values from average strength tests may run as high as 50 per cent, and even higher in pieces of large size. This marked variation is generally due to some inherent weakness of the individual piece such as a partial fracture or internal cavity that is not apparent from a surface examination. From this it is evident that whenever possible it is well to test each article at some loading greater than that to which it will be subjected in use.

It is apparent for the same reasons that clay parts are not resistant to sudden impact loads as the stresses thus induced are again localized causing loadings over a restricted area which are greater than the ultimate strength of the ware.

Temperature Changes Dangerous

The average ceramic material is easily broken by sudden changes in temperatures, especially if the article is of large size. A number of new ceramics which are equal to cast iron in their ability to withstand thermal shock have however, been developed. In fact, it is virtually impossible to break them due to any thermal conditions encountered in normal use.

To join two pieces of ceramic material together there are but two practical methods. One is to

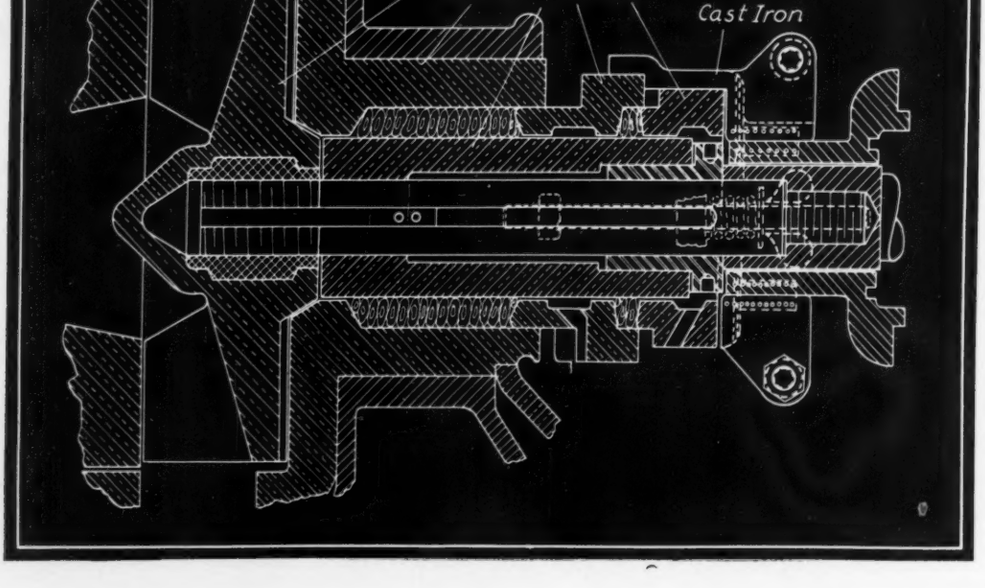

Fig. 2—All parts coming into contact with the fluid handled are of chemical stoneware

bind them mechanically by means of clamping devices and the other to cement them together. The former is the usual method and by far the stronger. If the parts are to form a fluid-tight seal they must be ground and clamped with a gasket between them, or lapped together. If lapping is used to accomplish a union the parts are not interchangeable, hence grinding and gasketing is the better procedure wherever possible.

There are few limitations to the size of parts that may be made from chemical ware. These may vary from parts having a volume of but a few cubic inches to those whose volume is in the order of 100 cubic feet.

In the chemical process and allied fields many rotating stoneware machines are in use. These include grinding mills, mixers, stirers, fans, centrifugal pumps, etching machines, etc. In these machines all parts coming in contact with corrosive mediums are of chemical ware.

TABLE I

Physical Properties of Chemical Stoneware

Max. ultimate tensile strength, pounds per square inch	7500
Max. ultimate compression, pounds per square inch	117,000
Max. ultimate bending strength, pounds per square inch	14,000
Max. ultimate strength in torsion, pounds per square inch	4600
Modulus of elasticity, pounds per square inch	5.96×10^{6}
Coefficient of thermal expansion per degree Fahr.	0.083×10^{-6} to 2.4×10^{-5}
Thermal conductivity B.T.U. per square foot load per degree Fahr. per foot per hour	0.8 to 26.4

In order to follow through the design of a specific machine it might be well to show the construction of an acid pump of recent design, *Fig.* 1. By so doing it is possible to show how advantage of the positive qualities of clay material has been taken and the negative qualities minimized. The best proof of the ruggedness of chemical ware is given by the fact that these pumps have been made in sizes that require over 30 horsepower.

The design in most general use is shown sectionally in *Fig.* 2. Two types are made, one with separate bearing pedestal and one with integral motor. The design of the parts involving stoneware is identical. The drawing shows an impeller of the open nonclogging type although they are also made with enclosed double shrouded impellers. All parts coming in contact with the fluid handled are of chemical stoneware and hence proof against all acid or salt corrosion except that induced by hydrofluoric acid or salts of that acid. All the parts are easily replaceable as they are all held by clamps rather than cement and are ground to finish tolerances to make them interchangeable. Tolerances on all parts are held as close as in good metal practice. An outside armor of cast iron is provided

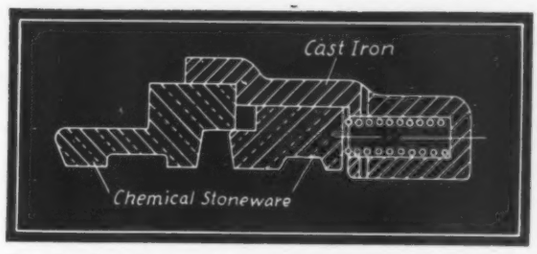

Fig. 3—Gland shroud sleeve is made of wear resistant ceramic material

to protect the chemical ware parts from rough treatment or impact shocks.

The chemical ware parts involved are the casing, casing cover, impeller, gland shroud sleeve, two gland followers and gland drain cup. Certain design features of each part are of interest as they illustrate some of the general comment made in the foregoing.

The internal casing of stoneware is lathe ground on all external surfaces which come in contact with the cast iron case. Likewise, internal parts of the iron case that touch the stoneware are machined. This gives the required intimate and strong union between clamped members. Internal areas of the stoneware are all finished by grinding with the exception of the volute and discharge pipe. (Actually the pumps do not have a volute decelarating chamber but have chambers of constant cross section of the Shearzer type which was found to give a higher efficiency for this general design). All clamping stresses on the case are compressive stresses, which in part compensate for the tensile stresses induced by the internal fluid pressure.

The recessed ring clamp provided for holding the suction line seats on a carefully ground groove in the stoneware, and the cast iron cover clamp that holds the stoneware cover in place is machined on the surface that touches the cover. A gasket is used between the case and cover. The impeller is ground on all surfacs except the fluid passages. These are glazed and offer a low resistance path.

Impellers Balanced and Tested

The center of the impeller is recessed and in this space a special alloy metal is poured, which is bored and threaded, and a keyway cut. The impeller is balanced by grinding small notches from the circumference of the rear shroud wall. These impellers are tested at 4 times their normal centripital load by running them at double operating speed after balancing. Normal running clearance between case and cover is 1/64 of an inch.

Of utmost importance in any centrifugal acid pump is the sealing gland. Design of the gland is of particular interest in these pumps as it illustrates a method of joining two dissimilar

(Concluded on Page 80)

SCANNING THE FIELD FOR IDEAS

NEW PROCESS CUTS COST

CONCRETE evidence of the employment of an idea aids the visualization of its application to other fields. Studying new developments in cameras for instance, brings to light several features that might be used to advantage by designers of other types of units.

From a materials standpoint with particular attention to processing, the small camera shown at right, *Fig.* 1, is noteworthy. Selling for only thirty-nine cents, this unit is manufactured of

Fig. 1—*Two significant factors in design—style and the possibilities of plastic molding—as exemplified by new cameras*

textolite selected for its lightness in weight and tensile strength. The body is made in three molded parts and all essential metal parts are rustproof. Because tolerances on the molded parts are well controlled, no machining is required and perfect fit allows no leakage of light.

There also is in production a bakelite molded camera. Like the unit discussed in the foregoing, it is durable under varying climatic conditions. Motion picture cameras have for some time been manufactured of this material because of the extreme changes in temperature and humidity under which they are used. This application of bakelite was a forerunner of molded camera cases of all types.

Styling has gone hand in hand with new processes and materials. One excellent example of this trend is the aerial camera, left, *Fig.* 1. Early models had a plain round cone. Now the cross section of the cone is approximately octagonal, from the magazine end to a point near the rim where it is faired into a circular cross section of smaller diameter. The effect is enhanced by the use of chromium strips.

Utility of the camera, in general, has tremendous possibilities. High speed motion picture photography to study movements of machine parts and mechanisms is increasing. By being able to slow down photographically the operation of the machine and observe what takes place, difficulties frequently are overcome and corrections made by redesign. Baron Chisaburo Shiba has developed apparatus capable of recording pictures at the rate of 20,000 frames a second (M. D., March, 1930). Supplementing this is a recent development recorded in the *New York Times,* of a motion picture camera with a speed of 6000 pictures a second, and even 80,000 under certain conditions. Inventor is R. Thun, a German engineer. The mechanism operates so rapidly that the film is dropped into a lightproof bag to save time and mechanical winding difficulties.

BALANCED DIAPHRAGM STOPS NOISE

MECHANICAL features of a new check valve show original ideas. The diaphragm is made of a composition, Thiokol, impervious to oil and unaffected by water. By balancing this diaphragm and designing the arrangement so

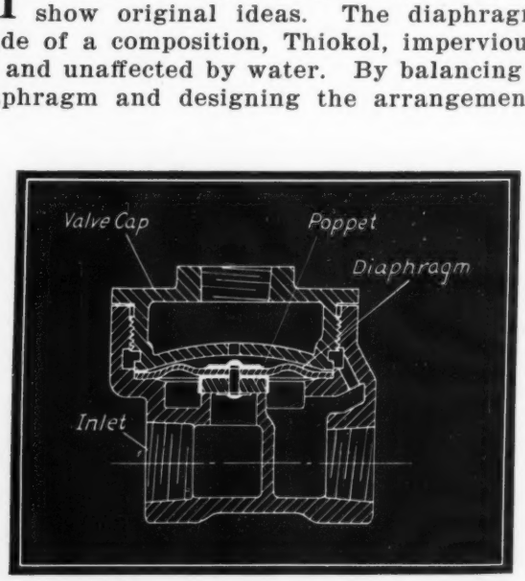

Fig. 2—*Diaphragm of composition material is balanced by pressure in this check valve*

that the poppet which is attached to it is suspended while in operation, noise is eliminated. Moreover, there are no close clearances where a film of oily sludge would cause the valve to stick.

The diaphragm is held in place by the shoulder of the valve cap, *Fig.* 2. When the cap is screwed into place the valve poppet barely touches its seat. Holes are provided through the cap to permit air from the tank or discharge side of the valve to act on the top of the diaphragm. The valve body around the seat is cored out so that the pressure from the discharge or tank side of the valve can also act on the bottom of the diaphragm. Thus, with the tank pressure on the top and bottom of the diaphragm, a balance is gained.

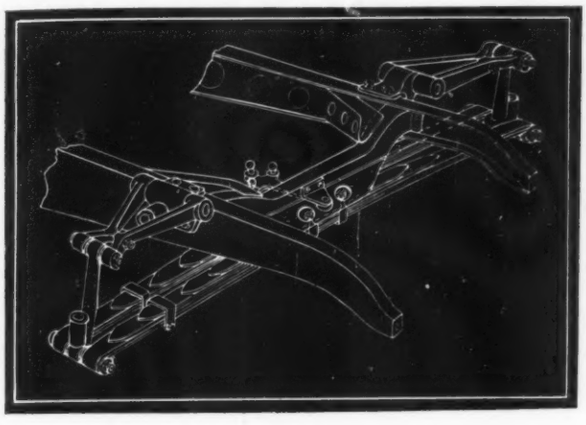

Fig. 3—*Front axle of auto is eliminated by new leaf spring front wheel suspension design*

As the diaphragm is in balance constantly and as the incoming air sets only on the poppet and not on the diaphragm, danger of diaphragm failure is eliminated. When in operation the valve poppet is raised slightly off its seat and remains in a practically suspended position. If the compressor is stopped the back pressure continues to act on both the top and bottom of the diaphragm. However, as the area exposed to tank pressure on the bottom of the diaphragm is less than that on top, the poppet is forced down on the seat, the entire pressure load is taken by the poppet and the diaphragm remains with equal pressure on top and bottom. The valve is manufactured by Milwaukee Valve Co.

LEAF SPRING PERMITS AXLELESS DESIGN

COIL springs were prominent in independent springing designs for 1934 model autos. Leaf springs seemed to be left out of the picture. But indications are that tranverse springs are equally as adaptable to independent springing. The Leaf Spring institute, Detroit, has developed a system, *Fig.* 3, that is said to reduced both the total and unsprung weights. In addition to features shown in the illustration, spring eyes are double wrapped as an additional factor of safety. Spring stiffness is designed to increase progressively with deflection through the use of pressed steel spring seats which shorten the effective length of the spring under load, as is done in some instances in heavy duty truck design.

ELIMINATING THE STUFFING BOX

MACHINES required to process that most unstable of materials, everyday dairy products, require a detailed attention to design far in excess of that usually encountered in routine problems. This is due to the necessity for the machine not only to meet its physical requirements but also to meet the most exacting requirements of health boards.

An instance of such a machine can be found in the design of a pasteurizer used for processing cream, commercial buttermilk, etc. In this unit it is necessary to secure proper circulation of the product without undue and damaging turbulence or concussion. In addition, there must be assurance that all portions of the dairy product being treated would not merely swirl around but are caused to contact efficiently the heating and cooling coil.

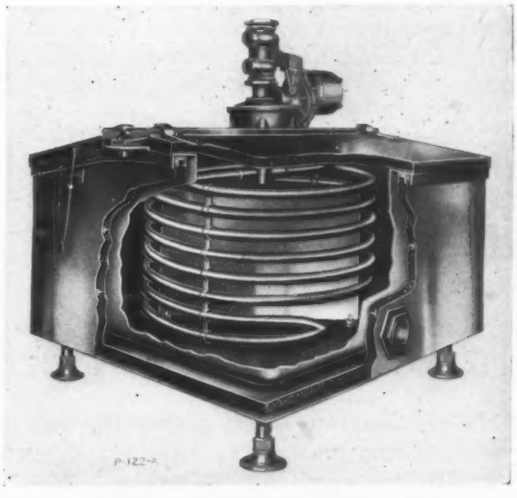

Fig. 4—*Complete contact with heating coils is obtained by arrangement of baffle plates*

Circulating coils early were employed to heat and cool dairy products, but the liquid in the center of the container (which would also be in the center of the coil) would not rotate properly. Consequently the next step was to adapt a baffle plate to the shaft, which thus became a part of the coil during its rotation; but again in this case a considerable portion of the liquid —that is, that part adjacent to the coil and its revolving baffle plate—was not properly stirred.

A disadvantage of early designs was the neces-

sity of sealing and lubricating the shaft at the bottom of the vat to prevent the escape of cream, but such stuffing boxes as were used proved exceedingly unsanitary and undesirable. These disadvantages were eliminated by the design of a patented revolving vertical coil pasteurizer, *Fig.* 4, which makes use of a stationary baffle located at the center of the machine, within the coil. The baffle is attached to the bottom during operation of the machine but removable for cleaning purposes. Stuffing boxes or shaft connections in the bottom of the vat have been eliminated by supporting the coil from an overhead bridge. Weight of the coil—that is, its thrust—is taken by thrust bearings, and proper seal is provided between the bridge and the shaft to prevent the lubricant or foreign materials of any kind from falling into the product being treated.

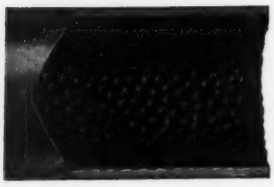

Fig. 5—By knurling the inside surface of conduit pipe, friction is minimized between the contact surface and the cable being pulled through it

Sanitation in this unit, which is built by the Jensen Creamery Machinery Co., is of primary consideration. In addition, the machine is made resistant to destructive action of milk acids by the use of 18-8 stainless steel wherever the dairy product comes in contact. The vertical coil is of welded stainless steel or Inconel tubing, goosenecks at the return point being formed in such fashion as to reduce concussion as a result of the coil revolving through the dairy product during treatment.

A LOCOMOTIVE WITHOUT FIREBOX!

STORING heat energy in water for use in the form of steam away from the changing point is a unique although not an entirely new idea. One recent application is the Heisler fireless locomotive, *Fig.* 6, which has no boiler or firebox. In place of the boiler it has a welded tank with a capacity of about three times that of the ordinary locomotive boiler. This tank, heavily lagged and jacketed to prevent loss of heat, is filled with water to about four-fifths of capacity. By a steam pipe connected from a stationary boiler to a point below the level of the water in the tank, the liquid is heated until the pressure and temperature in the locomotive tank are the same as in the stationary boiler from which the charge is taken.

Energy stored in the locomotive tank in the form of heat is transmitted to the rails as tractive force through the expansion of steam in the low pressure cylinders of the locomotive. After charging there is of course steam in the space above the water in the tank. As this steam passes through the throttle in small quantities and is expanded in the low pressure cylinders as the locomotive works, the pressure and temperature of the steam remaining above the water in the tank are reduced. During this reduction part of the water is turned into more steam, this process continuing until atmospheric pressure is reached.

The engine can be charged at any stationary boiler which carries a pressure of 100 pounds or more. One charge may last from two to ten hours under normal conditions of service, depending on the amount of work required of the locomotive. Because steam is generated far more economically in stationary boilers than in locomotive boilers, the idea is particularly attractive. This is believed to be the first locomotive to use a fusion-welded tank.

KNURLED SURFACE REDUCES CONTACT

KNURLING of the inside surface of conduit pipe has been found an excellent idea for minimizing the internal friction between the conduit and the cable being pulled through it. This definitely refutes the belief that the smoother the inside of the conduit the easier cable goes through. Engineers of Steel & Tubes Inc., Cleveland, have worked out the knurled design, *Fig.* 5, that prevents the cable from contacting on a large area. Instead of being in continuous contact with the wall of the conduit, the cable travels over the top of the small, round raised knobs. Tests show a reduction of 30 per cent in the effort required to pull the cable. Processing of the conduit surface is carried out prior to forming and welding.

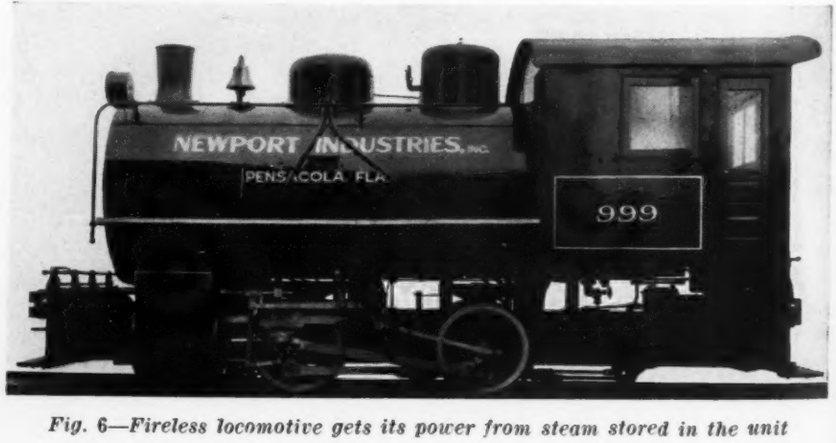

Fig. 6—Fireless locomotive gets its power from steam stored in the unit

Hard Facing, Inserts and Plating Satisfy Hardness Requirements

By Allen F. Clark

AS IS often the case with outstanding developments, applications have been found for cemented tungsten carbide that are far removed from its original field of machining and wire drawing applications. One of the most important new developments of this material—a development which has had wide application during the past year—is its use as a wear resistant medium on machine parts subject to rapid abrasive or corrosive wear.

As pointed out in the article on page 27 of the May, 1934, issue of MACHINE DESIGN, a satisfactory and simple method of designing parts which must withstand considerable abrasive action, and yet a method that will engender user satisfaction without unduly increasing costs, is to treat the surfaces exposed to the wear so that frequent replacements are unnecessary. The application of hard-facing rods was considered in May. The following will deal with the use of cemented tungsten carbide inserts, while the use of chromium plating for wear resistance will be covered in a forthcoming issue.

New Metallurgy Involved

Cemented tungsten carbide is composed of extremely fine, hard particles of tungsten carbide cemented together with a bonding material. It involves a comparatively new metallurgy which has resulted in extreme density. Not only is it also the hardest man-made metal, being next to the diamond in hardness and rating, 91 Rockwell C, but it has the highest compressive strength of any commercial material. The usual method of application is to braze a small blank of tungsten carbide in the machine part at the point of wear. Blanks are supplied in practically any shape desired, some of which include conical inserts, external and internal rings, square, curved and special-shaped inserts. The user may obtain the tungsten carbide blanks and do the necessary brazing and finishing or he may obtain the completely finished part from the manufacturer of the wear resisting material.

Tungsten carbide is particularly suitable for

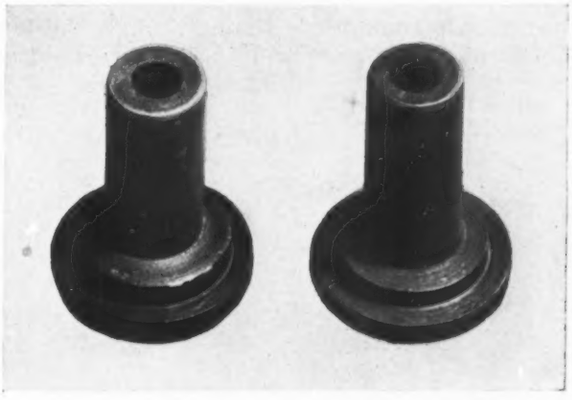

Fig. 1—Inserted rings of tungsten carbide materially increases nozzle life

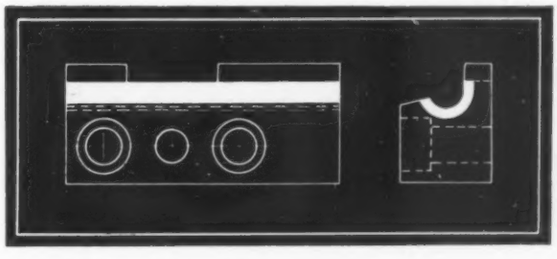

Fig. 2—Grinder rest employs an arcuate piece of wear resisting material

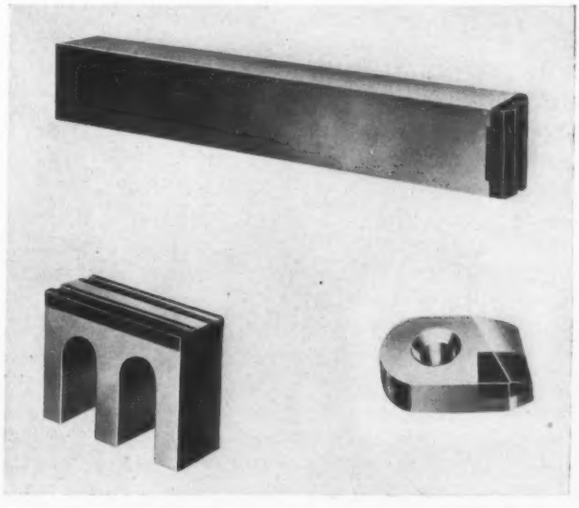

Fig. 3—Coiling points and coiling guides employ inserts advantageously

parts where replacement involves a considerable amount of down-time. On such applications the saving is not only in the cost of the part but also in a large reduction in labor. The material also finds many applications in the corrosion resistant field as it is resistant to most acids.

When applying tungsten carbide, it is well to remember that due to its extreme hardness it may have an abrasive effect on any part of the machine that contacts it regularly, yet is not so protected. To obtain best results, the contacting part should be similarly equipped at the point of contact. Another point which the designer should consider is that although the material can hold up under a fair amount of shock,

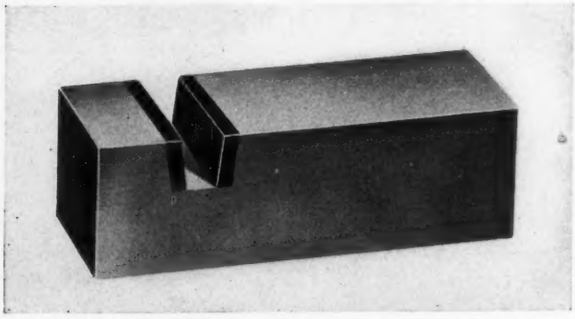

Fig. 4—Two blanks resist the wear encountered in parts used for the guiding of wire

as demonstrated by its use for intermittent cutting where impact is often quite severe, it cannot withstand extreme shock, and therefore cannot be used on applications where both abrasive wear and shock combine to break down the part. Another consideration, in applying the material to large areas, is its higher cost as compared to other metals used for wear resistant purposes. However, as most applications require only a small insert of the material at the exact point of wear, the increased life of the part, which conservatively is 10 to 25 times greater than that of ordinary metals, will in most cases offset the slightly higher cost.

Finds Wide Application

Machine part applications of cemented tungsten carbide that are now in successful use include hydraulic press valve stem and seat inserts; guides for wire, paper products, threads, leather, steel parts being processed in welding machines, etc.; nozzles for all types of abrasive materials such as welding rod coatings, soaps, and similar substances; chip breakers; master cams; grinder and lathe rests; and many others.

A typical wire guide is shown in *Fig.* 4. In this part two blanks are used, one on either side of the groove through which the wire runs. A coiling point used in the wire industry which makes use of a grooved piece of the material brazed into the shank metal is shown at the top of *Fig.* 3. The piece at the lower left in this illustration is a coiling guide for the same industry which employs a piece which has a semi-circular groove.

An interesting problem solved by use of the material was developed in the soap industry where liquid soap under 1200 pounds pressure was blown through a nozzle. Trouble was encountered with the nozzles wearing rapidly and particles thus wearing off effecting the specific gravity of the soap so that the full weight of soap would not fill the box to the required level. The development of the insert shown in *Fig.* 5 solved this problem as the material's extremely slow rate of wear did not effect the specific gravity of the soap. A successful application of inserted tungsten carbide on a sand blast nozzle where extremely severe wear is encountered, is illustrated in *Fig.* 1 The nozzle at the left is the piece before it had been used, that at the right is the piece after twenty hours wear. The productive life of these nozzles is 60 hours, compared with a former life of about one-half hour.

A typical tool rest which has been redesigned to make use of the inserts is shown in *Fig.* 2, a grinder rest used in grinding the angle on the

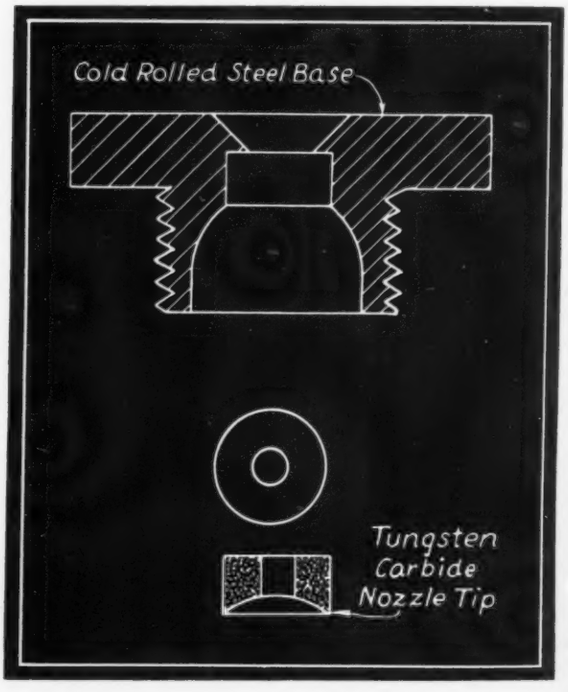

Fig. 5—Specially shaped tungsten carbide inserts solves erosion problem in soap nozzle

head of a valve. The piece it replaced had to be changed every few days while the redesign gave many weeks of service.

These are but a few representative applications of cemented tungsten carbide in this field. They may, however, serve to suggest to the designer a solution to his problems of rapid wear.

For assistance in the preparation of this article and for the illustrations used, MACHINE DESIGN wishes to acknowledge the assistance of Carboloy Co., Detroit.

Practical Aspects of Spring Applications

By E. W. Weaver

WHEN the first mechanic coiled a wire or rod in a helical manner he started a line of action only a little less useful than the originator of the first crude wheel. Now every designer of machinery employs the art, if not the science, inherent in the many ramifications of this type of construction.

The most common practice embodies the use of a spring to exert tension or compression in an axial direction. Another involves its use as a spring—so-called torsion—in which there is a tendency either to wind up or unwind. Or still another example is a spring as a thread, either external or internal.

In the close wound, small diameter helix it becomes the common flexible shaft having, of course, a contra-wound spring fitting snugly within the outer coil, with the ends anchored together. A powerful friction clutch results when spring wire is wound either on or within a cylindrical surface. We might not recognize the science in the lowly spring lock washer, but will notice its adaptability to pencil and card holders and to electrical resistance coils.

Some of the most interesting and useful applications of springs take advantage of more than one phase of their capabilities. For instance, the use of a spring, *Fig.* 1, right, in a well known electric gasoline pumping unit brings out some of its possibilities. This design furnishes a means of adjustment for regulating the by-passing action of the pump when the nozzle valve is partially or entirely closed and the pump is still running. It compensates for the varying requirements of blade follower in the double cam type of pumps. The unusual feature of application consists in the use of a spring as a floating bearing for the rocking pressure bar holding the by-pass blades in engagement with the cams.

Inasmuch as the lower end of the spring is piloted rigidly on a cylindrical lug integral with the centrally located rocking bar, there is no

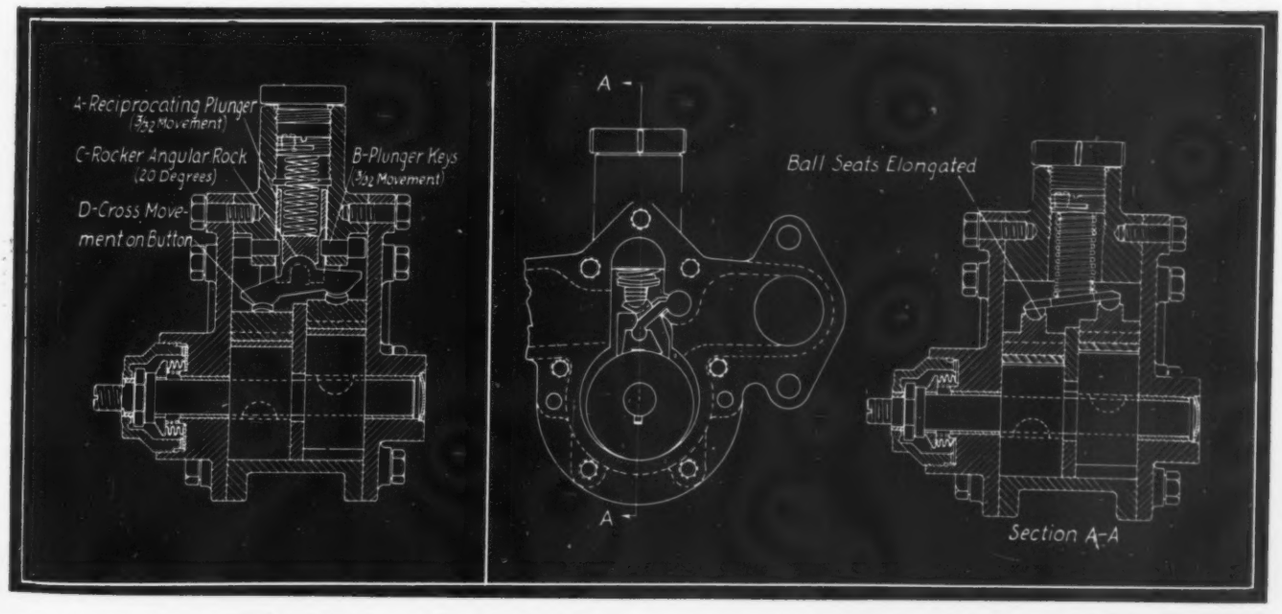

Fig. 1—*The spring in the mechanism at the right acts as a floating bearing and makes* 3500 *flexes per minute. In this design, succeeding that at left, parts A, B and C have been eliminated, thereby increasing efficiency and obviating noise*

pivot bearing and consequently no sliding friction. It is necessary for the spring to flex back and forth at each oscillation of the bar. This means two flexes to each revolution of the pump shaft. Angular motion for each "rock" is about 20 degrees. As the pump shaft ordinarily is driven by a motor running at 1750 revolutions per minute, there are 3500 complete flexes per minute.

In working out the construction, the designer of this spring admits to a certain amount of misgiving as the spring was already stressed to about 80,000 pounds per square inch at maximum pressure adjustment. However, as is sometimes the case, a situation had developed in pumping "gummy" gasoline that required change. With the previous construction, left *Fig.* 1, a closely fitting plunger of good length, keyed to prevent rotation in its cylindrical hole, carried the rocking bar in an ordinary cylindrical half bearing. When these closely fitted bearing surfaces became gummed up, noise resulted and action was faulty. Consequently the mechanism was redesigned as shown at right. *Fig.* 1. It will be noted that the parts indicated at *A, B,* and *C* in the left-hand drawing were eliminated.

Fig. 2—A double-duty spring employed in an auto starter drive acts as a thread to produce end travel of pinion

Life of Spring Compared

After it had been definitely proved that the spring would function satisfactorily in the manner described, it was necessary to determine whether the life of the spring under these conditions would compare with the life of other replaceable parts. As this type of equipment comes in the infrequently used class, pumping possibly two hundred gallons per day in a total of probably twenty minutes of operation, the conclusion was reached after certain test runs that it was entirely feasable. Moreover, the expected life of the unit from an obsolescence standpoint is five years.

It is pleasing to add that oil company breakdown tests proved conclusively its entire suitability. Tests were equivalent to seventeen years of operation, and beyond a slight polishing at a point about one-third of the length of the spring from the lower end, where the barrel of the spring contacted with the smooth clearance hole in which it worked, it appeared exactly as when the test started. The number of flexes was computed to be around 525,000,000. Further, in two years of operation of several thousand of these pumps, there is no record of spring failure.

Another interesting double duty spring is that used in the automobile engine starter drive made by the Burgess-Norton Co., Geneva, Ill. While nearly everyone is familiar with the Bendix mechanism in which the starter pinion of the motor shaft travels axially on a screw-threaded shaft, not so many are acquainted with the Charter Drive made by the previously mentioned firm for the same purpose.

Spring Actuates Pinion Endwise

In this drive, shown in *Fig.* 2 the pinion is mounted on a smooth sleeve and advances axially into engagement with the flywheel gear, using the spiral of the spring *Fig.* 3, as the actuating means for obtaining end travel. A sleeve or "driver" with a tongue or "tracker" projecting from the cylindrical section is keyed to the armature shaft and rotates with it. The projecting "tracker" fits loosely between adjacent turns of the spring. When the current is switched on the armature shaft starts to rotate, and as there is an inertia drag to the pinion and its driving spring, there is a relative movement between the driving tongue and spring helix. In this manner the pinion is forced axially along the sleeve into engagement with the flywheel gear.

Whenever the teeth instead of meshing properly butt against each other the spring will give somewhat and allow the pinion to rotate sufficiently to engage. As the engine fires and becomes the driving element the speed of the pinion becomes much greater than that of the armature shaft, reversing the axial movement and disconnecting the pinion from the flywheel gear. The other duty of the spring is, of course, to act as a drive cushion through the torsional elasticity of the helically wound, multi-coil spring. In this it functions exactly as does the spring used with a Bendix drive.

Fig. 3—In this spring both ends of the wire have been formed to serve as driving lugs

Use Governs Choice of Gear Steels

BECAUSE no one steel or treatment is suitable for all applications, the designer of machinery is faced with the problem of selection of suitable material for a specific use. The solution to some of the complexities of the situation is found in data on "Steels for Heat Treated Gearing," compiled by T. R. Rideout of Westinghouse. He discussed this subject before the recent annual meeting of the American Gear Manufacturers association.

Adoption of heat treated gearing has become necessary to meet the demands of increasing loads and speeds and the economic requirements of longer life. Qualities which a steel should possess in order to measure up to service and manufacturing requirements were named by Mr. Rideout in the order of their importance as follows: Ability to withstand wear; strength; static toughness; impact toughness; low amount of distortion; ease of machinability; suscepibility to and ease of heat treatment.

The reason for listing ability to resist wear as of primary importance is apparent, he asserted, when it is considered that properly designed and correctly treated gears give far less trouble from tooth breakage than from rapid wear. Moreover, the fact that so little is known regarding the relative wearing qualities of the different heat treated steels makes it of utmost importance.

Discussing strength, he said that contrary to popular opinion the endurance limit does not vary with the composition of the steel but only with ultimate strength. This latter factor varies only with the hardness of the steel which is purely a function of the carbon content and heat treatment. For instance, an oil-quenched alloy steel might have the same hardness as a water-quenched plain carbon steel of the same carbon content. Similarly, a low carbon oil-quenched alloy steel might have the same hardness as a higher carbon content oil-quenched plain carbon steel. For the same hardness any of these four steels will have, within close limits, the same ultimate tensile strength and endurance limit. Endurance limit for treated steels having a martensitic structure will be 40 per cent of the ultimate tensile strength, whereas for a pearlitic structure it will be 45 per cent. This applies to polished plain test specimens only. The values vary if there are stress concentrations due to grooves or fillets.

Use of untreated alloy steels for gears is seldom if ever justified and then only when heat treating facilities are lacking. It is true that the strength and toughness of normalized alloy steels are somewhat greater than for an untreated plain carbon steel of the same carbon content, but in Mr. Rideout's opinion the higher values of those properties alone do not justify the additional cost, and in no case will an untreated or normalized alloy steel be comparable with a heat treated plain carbon steel.

Selecting the Type of Steel

Factors which determine whether to use heat treated plain carbon steels or heat treated alloy steels are: Does the service condition or design require the superior characteristics, or if alloy steels are not required, will the advantages to be derived offset the additional cost? For most applications, Mr. Rideout contended, plain carbon steels, heat treated to obtain the best of their qualities for the service intended, are eminently satisfactory and quite economical.

Another point emphasized is that there is con-

GEAR technology, standardization, merchandising, costs and employe compensation were highlights of the eighteenth annual convention of the American Gear Manufactuers association held recently. At the concluding session code and economic matters received attention. With business among the gear makers during the first quarter of 1934 almost 100 per cent improved over the corresponding period of 1933, a highly optimistic outlook characterized the meeting.

siderable difference of opinion regarding the ability of case hardened steels to withstand shock load successfully. Due to the tendency of the extremely hard and brittle case to crack or spall under shock loads, Mr. Rideout prefers full hardening steels for such applications. In order to realize the greatest benefits from the core properties, case hardened steels should be double quenched. This is particularly true of alloy steels because the benefit derived from their use seldom justifies the additional expense unless the core is refined and toughened by the second quench.

Obtaining Decreased Distortion

Decreased distortion can be obtained by employing low quenching temperatures, rapid heating, and slow cooling at quenching. Low quenching temperatures can be had by the use of some of the alloy steels which have low critical points, principally the nickel and molybdenum steels. Slow cooling at quenching is obtained by the use of an oil bath, or for some of the less frequently used air hardening steels, by gradually cooling the air.

Machinability of a steel, like its wearing qualities, is a function of the hardness and toughness of the material. On gears which are machined before heat treating, good machinability is obtained by normalization. The hardness at which machining after heat treatment is practical and economical varies considerably with the type of steel, the treatment and the amount of hardness penetration. In general, the qualities which lower the allowable machining hardness are a fine grain size and a high degree of toughness. Some of the alloying elements tend also to make machining difficult.

Lists Characteristics of Alloys

For use in deciding on a particular kind of alloy steel, Mr. Rideout enumerated in his paper some of the more important characteristics imparted by the different alloying elements.

Additional data covering gear material was embodied in the report of the metallurgical committee, Chester B. Hamilton, chairman. The report covers a series of tests run by his company, Hamilton Gear & Machine Co., Toronto, Ont., to compare the load bearing abilities of different materials for worm gears. From results obtained it appears that nickel phosphorus bronze, chill cast, was the best of the materials tested at all speeds. Its analysis is copper 87½, tin 11, nickel 1½ and phosphorus 0.1 to 0.2 per cent. Apparently, Mr. Hamilton asserted, it is detrimental to add nickel unless the casting is made in a chill, but an advantage if chilled. Navy bronze containing 2 per cent zinc, no phosphorus and not chilled, is equal to sand cast nickel bronze at all speeds above 600 revolutions per minute, but lacks strength for high torques at lower speeds. Manganese bronze and aluminum bronze not only failed at low loads but in doing so they both destroyed the worm, which was not the case with any of the other gear material tested.

Plans have been completed for new publications to be sponsored by the gear association. One is a book containing gear data, with Prof. Earle Buckingham, Massachusetts Institute of Technology, as editor. The second book which probably will be combined with the first, will contain A.G.M.A. standards. Gear tables will make up the third publication.

The keyway committee headed by R. B. Zerfey, Tool Steel Gear & Pinion Co., Cincinnati, will extend the scope of its activities to include the standardization of involute splines. In his report, the chairman outlined the situation that exists regarding standardization of straight keys, plain and gib head tapered keys. Members voted to follow ASA practice in this connection.

Contrasting the improved status of business in the gear industry today with that of a year ago, President E. W. Miller, Fellows Gear Shaper Co., Springfield, Vt., opened the convention. The beginning of 1933 did not bring much encouragement, he asserted, but now business is better. Statistics of the gear industry show that there was a little improvement before the middle of 1933, the total for that year being 50 per cent better than that of 1932. Of the entire business for 1933 more than 60 per cent was done in the last half of the year. Mr. Miller this year relinquishes the presidency of the association to John Christensen whose biographical sketch appears on page 51 of this issue.

OF outstanding importance is the contribution on gearing commencing on page 40. This treatise, which is being presented in two parts, covers a comprehensive analysis, written exclusively for MACHINE DESIGN, of the load-carrying capacity of helical and herringbone gears. W. P. Schmitter, the author, includes in it much of the data given in his paper, "Helical and Herringbone Speed Reducer Ratings," presented at the A.G.M.A. annual meeting.

Design Given Increased Attention

New Books Cover Fundamentals of Designing and Plastic Molding

Machine Design

By Louis J. Bradford and Paul B. Eaton; published by John Wiley & Sons Inc., New York; available through MACHINE DESIGN *for* $3.00 *plus* 15 *cents postage.*

In preparing the third edition of this text book the authors have attempted to bring the work abreast of developments in this field of machine design without materially increasing its size. Principal changes have been made in that portion which deals with lubrication, and charts have been substituted for much of the material relating to bearing design contained in the first two editions. These charts permit accurate design, without recourse to difficult mathematics, of bearing to which the hydrodynamic theory applies.

Because of the increasing importance in welded construction, a section dealing with this phase of engineering has been included. Additional problems have been provided, some without answers, in order to increase the usefulness of the text to teachers of the subject. Those who are seeking an understanding of the fundamentals of design will find this book helpful.

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Design and Use of Instruments and Accurate Mechanism

By T. N. Whitehead; published by Macmillan Co., New York; available through MACHINE DESIGN *for* $3.50 *plus* 15 *cents postage.*

Although written primarily for designers of instruments and for those who use them, the subject matter of this volume transcends these limits. Some of the principles involved are equally applicable to mechanism as a whole; still more are pertinent to a large class of light machinery which has many of the essential characteristics of an instrument.

In this book the examples chosen will in many cases show the similarity or identity of the more general principles of machine design. As the author brings out, the designer must rely less on detailed knowledge of particular examples and special devices than on a body of underlying principles which will serve to guide him in novel situations.

The volume represents a conscious revolt against mathematical abstractions when unaccompanied by an independent examination of the actual facts. Its aim is to concentrate on those parts of the subject which seem to call for revision and with which designers of mechanical units are likely to be less acquainted. The theory of errors is covered in the first part and discussion on various phases of design occupies the second section.

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Plastic Molding

By Louis F. Rahm; published by McGraw-Hil Book Co. Inc., New York; available through MACHINE DESIGN *for* $3.00 *plus* 15 *cents postage.*

In designing parts to be made from plastics the engineer must necessarily know something about the materials, equipment and methods employed in production. This book is written as an introduction to those phases of the subject of plastic molding. The author sets forth that the most remarkable development has come about in the manufacture of precise parts for mechanical and electrical uses. In these fields particularly, the phenomenal growth of the industry during the past twenty years has been due to the class of synthetic materials known as thermosetting plastics.

In compiling these data for the volume the author has attempted to present a general view of the methods and equipment employed. Stress laid on the molding of thermosetting plastics may be justified, he says, by the present importance of such materials and the probability that future plastics will be handled with substantially the same methods and equipment.

Concerning materials discussed, Prof. Rahm states that in view of the volume available on this phase of the subject detailed chemical data is unwarranted in a text of this scope. The four parts into which the book is divided cover molding process and plastics, molds, operating equipment, and the molding plant. The appendix includes an alphabetical list of plastic products with the names of the producers.

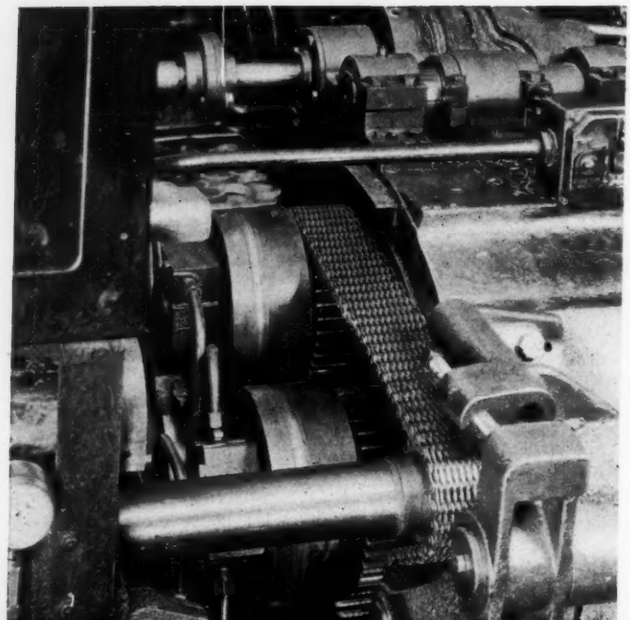


Fig. 1—Unusual six-spindle drive on automatic machine utilizes silent chain

Combating Severe Co Chain Drive

By L. E. Jermy

CHAIN speeds of 3000 to 4000 feet per minute are not uncommon today in power transmission practice. Formerly it was considered advisable rarely to exceed 1500 feet, the actual speed depending largely on the pitch of chain used. Given improved methods of lubrication, however, many drives of higher speed are operating satisfactorily.

Similarly with the maximum desirable ratios between driver and driven sprockets. A ratio of six to one for many years was and in fact still is believed to be the desirable maximum for a long life drive. Ratios of eight to one and ten to one can be employed however, and one case in particular is on record in which a 19.6-1 ratio was used successfully on a drive for intermittent service. But it should be kept in mind that these drives are of special nature and should under no circumstances be considered without giving the manufacturer of the chain an opportunity to make recommendations. The same thing applies to most of the drives discussed later in this article.

Roller or Silent Chain?

A question that often puzzles designers of machinery about to figure on a chain drive is whether silent chain or roller chain should be used. There is no specifically defined field for the use of one or the other though the tendency in the past generally has been to use silent chain for high speed, extremely quiet drives and roller chain for lower speeds. Since the introduction of short pitch and multiple strand roller chains however, these often are used also on high speed drives. Roller chain is somewhat more rugged than silent chain and consequently is capable of withstanding more severe conditions such as reciprocating or impulsive loads, dirty and moist locations.

Dirt and moisture obviously do not affect the decision as to which type chain to select in those cases where a totally enclosed and properly lubricated drive is to be installed. Consequently the choice is governed in most cases by consideration of all factors involved, with some preference in favor of roller chain where particularly severe or unsteady loads are to be transmitted and cost is of utmost importance.

One reason for holding down the ratio between

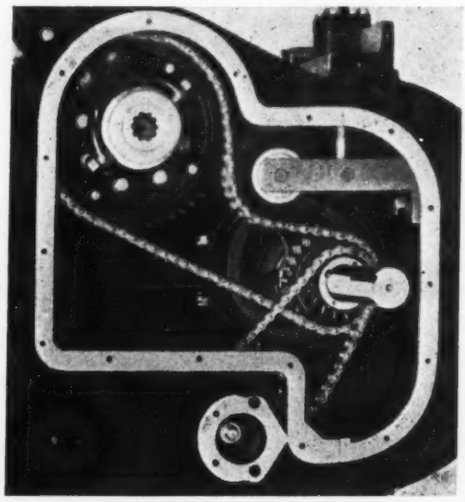


Fig. 2—Multiple strand roller chain affords compactness in front-end drive

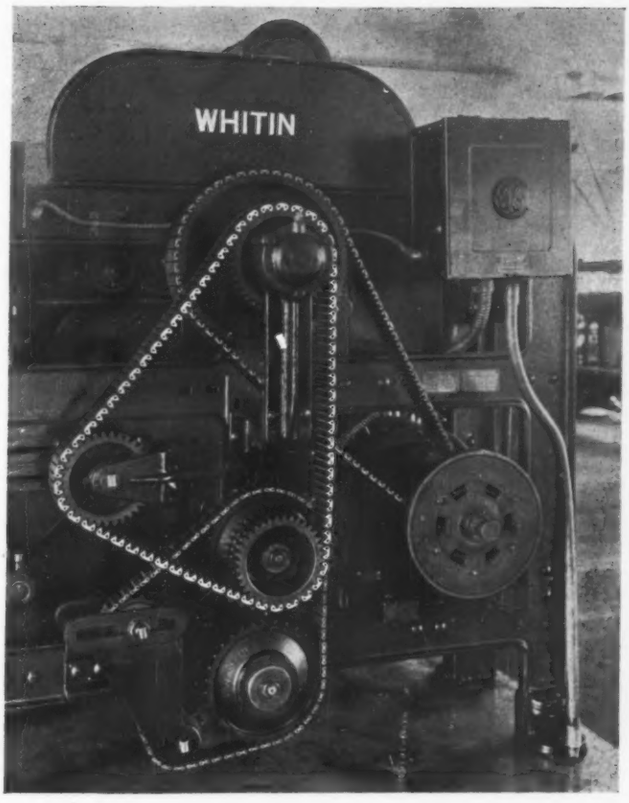


Fig. 4—Chain idlers are cut wheels, adjustable manually for take-up

Conditions by

the driver and driven sprockets is because long life cannot be expected from a high ratio drive. In the installation having 19.6-1 ratio to which reference previously has been made, the smaller sprocket having as few as 10 teeth was cut directly on the motor shaft, with a special tooth form on account of the small arc of wrap. The driven sprocket had 196 teeth. This drive used ⅝-inch pitch roller chain, ⅜-inch wide and was successful largely on account of the fact that it operated only at intervals. Ordinarily, with special sprockets having as many as 196 teeth, wear which is bound to take place in the chain would soon make replacement necessary because less chain elongation is permissible as the number of teeth increase. This factor is negligible on drives having not more than a desirable maximum of 140 teeth in the larger wheel.

Good practice in calculating a drive is to try to arrange for a pinion having not less than seventeen or nineteen teeth. Naturally, due to the polygonal action of the chain on engaging the entering teeth of a pinion, there is proportionately more "bend" between each link on a sprocket with 10 teeth—as discussed in the foregoing—or less, than with a sprocket that has more teeth. Smoothness of running and quietness, combined with positive action, that are obtainable readily with any well-designed drive for normal conditions therefore presupposes a pinion having a reasonable minimum number of teeth. If the ratio is too high to permit this it is desirable to arrange a double reduction drive.

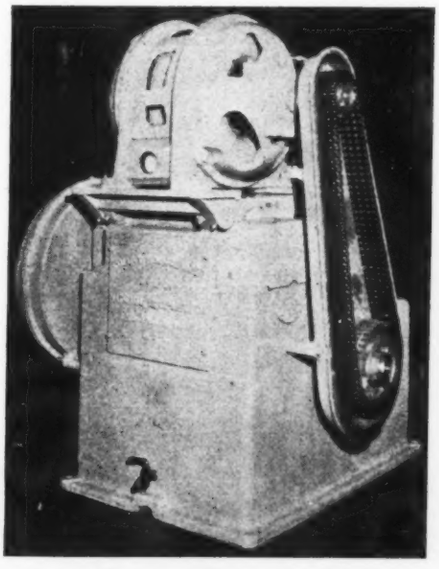


Fig. 3—Homogenizer is driven by quadruple-strand chain

Drives Six Spindles of Machine

Illustrative of a transmission of special character incorporated in the original design of a machine is the six-spindle drive on a Gridley automatic depicted in *Fig.* 1. This particular machine is used by ball bearing manufacturers in the machining of ball races each chucking head holding a race for a certain operation and then rotating for the next operation. The six rotating chucks are mounted in a single rotating head which is turned at the end of each operation independently of the chain drive. The chucking heads rotate within the chain while they are being moved from one operating position to the next, ceasing to rotate only at the loading station. In the illustration is shown the driving shaft with a chain idler operating on the back of the chain and three of the chucking heads with driving sprockets. Precaution has been taken to use a soft material for the peri-

phery of the idler, thus eliminating the possibility of damage to the backs of the chain teeth.

It usually is assumed that for a normal drive the arc of contact or wrap of the chain around the sprocket should not be less than about 120 degrees. If, however, as in the case just discussed sprockets have a fairly large number of teeth can be accommodated, thus allowing several teeth in each sprocket to be in contact with the chain, satisfactory results are obtainable. The Gridley installation is ample proof of this point.

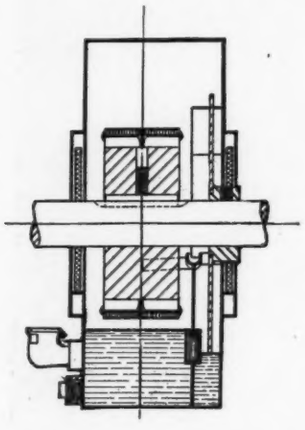


Fig. 5 — Lubrication problems are met efficiently by new design of oiltight chain-case. Note arrangement of felt seals, baffles and pipe for dripping oil on inner surface of chain

More normal transmissions are shown in *Figs.* 2 and 3. The former illustrates a diesel front end drive utilizing multiple strand roller chain for the drive to the camshaft. As indicated previously it is advantageous to use multiple rather than single strand chain in a restricted area of this type because to utilize single strand of the proper load-carrying capacity would necessitate going to a longer pitch than is feasible with the multiple strand chain. With the latter (made with as many as eight strands or even more for extremely special conditions) the sprocket diameters can be kept down and still be large enough to accommodate a reasonably high number of teeth. The figures in the table below are for single strand roller chain, and are based on standard practice for normal ratios between sprockets.

In *Fig.* 4 (chaincase removed) is shown a wool spinning frame drive using three silent chains two of which are adjustable manually with idler sprockets. Whenever practicable, idlers of this type are greatly to be preferred, as backward bend in the chain is obviated and increased life insured.

Employment of chain drives in which the chain functions in a horizontal plane with the sprocket shafts vertical is not encountered frequently, but a nationally known maker of dairy equipment has developed a line of bottle filling machines incorporating this idea for driving the revolving tables. Five sprockets are used wth one roller chain, the latter being lubricated by a pump. Chain and sprocket wear are insignificant in spite of the fact the chain weight is supported by the upper roller link plates. This is another instance in which a successful drive for peculiar conditions was devised with the aid of the engineers of the chain manufacturing company.

Acknowledgment is made to the following for their considerate assistance in the preparation of this article: Baldwin-Duckworth Chain Corp., Chain Belt Co., Diamond Chain & Mfg. Co., Link-Belt Co., Morse Chain Co., Ramsay Chain Co. Inc., and The Whitney Mfg. Co.

Articles on this and allied subjects published in previous issues of MACHINE DESIGN include:

"Adjustment Is Vital Factor in Chain Drive Performance," by Ralph Dyson, Oct., 1929, p. 35.

"Utilizes Chain with Cutting Edges," March, 1930, p. 20.

"Applying Silent Chain Drives," by J. H. Groom, Nov., 1930, p. 32.

"Indexing by Silent Chain," by Charles T. Plastow, Feb., 1931, p. 55.

"Use of Roller Chain Is Extended to Camshaft Drives," by Harold T. Shepherd, Aug., 1931, p. 39.

Horsepower Ratings and Working Loads for Standard Single Strand Roller Chains

Amer. Std. Chain No.	Pitch, inches	Width inches	Max. R.P.M. of Sprockets		Linear Speed of Chain in Feet per Minute									
					50	100	200	300	400	600	800	1000	1200	1500
35N	3/8	3/16	3733	HP	0.30	0.56	0.98	1.30	1.56	1.96	2.25	2.46	2.63	2.82
				WL*	197	183	161	143	129	108	93	81	72	62
41	1/2	1/4	3040	HP	0.36	0.67	1.18	1.58	1.90	2.39	2.73	2.99	3.20	3.43
				WL	240	223	196	175	157	132	113	99	88	76
40	1/2	5/16	2782	HP	0.49	0.92	1.62	2.17	2.60	3.26	3.74	4.09	4.37	4.69
				WL	328	305	268	239	215	180	154	135	120	103
50	5/8	3/8	1970	HP	0.78	1.46	2.57	3.43	4.13	5.17	5.92	6.49	6.93	7.44
				WL	520	484	424	378	341	285	245	214	191	164
60	3/4	1/2	1530	HP	1.18	2.20	3.86	5.17	6.21	7.79	8.92	9.77	10.44	11.20
				WL	782	728	639	569	513	429	368	323	287	246
80	1	5/8	970	HP	2.00	3.73	6.56	8.76	10.54	13.21	15.13	16.57	17.67	18.99
				WL	1326	1234	1083	965	870	727	624	547	487	418
100	1 1/4	3/4	682	HP	2.93	5.45	9.57	12.79	15.37	19.27	22.07	24.17	25.81	27.70
				WL	1934	1799	1579	1407	1269	1060	911	798	710	609
120	1 1/2	1	535	HP	4.42	8.22	14.44	19.29	23.30	29.08	33.30	36.47	38.98	41.80
				WL	2918	2715	2383	2123	1914	1600	1374	1204	1071	920
140	1 3/4	1	390	HP	5.29	9.85	17.30	23.12	27.83	34.85	39.91	43.71	46.68	50.08
				WL	3497	3254	2855	2544	2294	1917	1646	1443	1284	1102
160	2	1 1/4	327	HP	7.23	13.47	23.63	31.59	37.97	47.60	54.51	59.71	63.77	68.41
				WL	4777	4444	3900	3475	3133	2618	2249	1971	1754	1505
200	2 1/2	1 1/2	240	HP	12.21	22.72	39.88	53.30	64.08	80.33	91.99	100.8	107.6	115.5
				WL	8062	7499	6582	5864	5287	4418	3795	3325	2959	2540

*WL indicates working load in pounds.

Cites Functions of Alloying Elements

NEED has long existed for a philosophy of alloy steels which would provide a basis for more orderly and rational arrangement of the varied information on this subject. Even a scientific framework upon which might be supported, so to speak, the classified items of empirical knowledge would be a great help to the metallurgist, according to Edgar C. Bain, United States Steel Corp., speaking at the recent meeting of American Iron & Steel institute.

Evaluation, from the engineering standpoint, of the contributions toward useful steel properties made by the various elements, and their combinations, would be a most desirable service, but the inherent complexities of the problem preclude any hope of such an advanced state of knowledge in the near future.

Might Choose High Hardness

With a certain definite heat treatment a steel containing an alloying element may yield properties very different from those of carbon steel. However, if one wishes to inquire broadly into the overall advantages of an alloy steel, it is necessary to choose the optimum heat treatment for each of the steels, plain carbon and alloy, with respect to the desired properties. If, for example, the familiar combination of high strength and moderate ductility is the desideratum then one might choose a high hardness in the vicinity of 525 brinell and heat treat both steels to this condition and compare the tensile test results.

In *Fig.* 1 is shown the hardness distribution across the same section of three steels identically quenched. The increased alloy content of the fully hardened section accomplishes the necessary decrease of the critical quenching rate so that even the metal in the center, cooling most slowly, is still able to harden.

This is by all odds the most important contribution of alloying elements. Large sections may thus be rendered fully hard throughout, ready for tempering which develops the spheroidal carbide distribution conducive to maximum toughness for any given strength. All elements (with the possible exception of cobalt) even including oxygen, if they are dissolved in the austenite, appear to increase hardenability by influencing transformation rate.

Alloying elements function so as to accomplish one or more of the following:

1. Increase hardenability. Upon this effect depend the development of high strength throughout the whole of large sections, the development of higher strength by simple cooling and the overcoming of the shallow-hardening tendency of beneficial fine grain size. Manganese, silicon, chromium, nickel, molybdenum, tungsten, all contribute to deep-hardening by retarding the transformation rate in about the order named, but only in so far as they are in solution in the austenite at time of quenching.

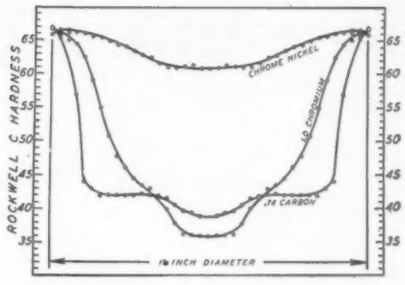


Fig. 1 — Deeper hardenability contributed by alloying elements — shallow, carbon steel; deep, chrome-nickel steel

2. Retain fine-grained austenite. Upon this effect depends not only to a great extent the hardenability of any single composition but, what is more important, the toughness of its harder structures. Grain growth is inhibited by the undissolved special carbides of vanadium, titanium, zirconium, molybdenum, tungsten, and chromium (somewhat in the order named). The oxide compounds of aluminum, zirconium, vanadium, titanium, and silicon if in persistently fine particles, likewise retard grain growth.

Ductility, Impact Strength Improved

3. Retard softening during tempering. Upon this effect depends the possibility of employing higher tempering temperatures for a given strength, lessening residual internal stress and improving thereby ductility and impact strength. The creep resisting property of a steel probably depends likewise upon this preservation of a fine dispersion of carbide particles. Red hardness is the result of this influence in high degree. The same carbide-forming elements exert this effect as was mentioned under (2), but only as dissolved in austenite and not as dissolved in ferrite.

4. Strengthen the pure ferrite matrix itself. Upon this effect of solid-solution hardening depends a method of moderate strengthening without loss of ductility. Nickel, silicon, manganese, cobalt, aluminum (as alloy) and possibly chromium function in this manner but only insofar as they are dissolved in ferrite and not incorporated in carbide.

Determining Capacity of Helical and Herringbone Gearing

By W. P. Schmitter

THE primary purpose of a gear horsepower formula of the type developed in this article is to assist the engineer in designing a set of gears for a particular requirement. The determination of the correct proportions for transmitting a certain load is a matter of vital interest to the designer, because if the fundamental basis is unsound, the gears will be either inadequate to carry the load, or so large as to place his company at a commercial disadvantage.

There has been a certain amount of confusion among engineers due to the absence of any conformity in the methods used. In one of the empirical equations in use the gear proportions are obtained from a consideration of the allowable pressure per inch of face width; in another, from face and diameter; in still another, from face and the square root of pinion diameter; and there are at least two others which are fundamentally strength formulas, but modified by the application of wear factors. The empirical formulas used by the different manufacturers have worked out quite successfully when applied to their own product and within the field for which they have been devised. In the hands of an experienced gear engineer they are good tools, largely because his background enables him to temper the arbitrarily chosen constants to suit any individual requirements. On the other hand, the general machine designer, without this specialized gear experience, is quite apt to obtain inconsistent and incongruous results.

The search for a method of rating which would give adequate consideration to the variations of design, tooth form, materials and heat treatment as expressed in the practice of the individual manufacturers led to the development of an equation having a rational foundation[1]. The purpose of this article is to direct the essential elements of that equation to the design of new gears rather than to the rating of existing designs. In addition, the range of application is extended to cover helical and herringbone gears in general. A listing of nomenclature begins on page 59.

The primary consideration in designing gears is to have tooth proportions such as will insure freedom from the usual types of surface and subsurface failures, namely, pitting, spalling, wear, scuffing, rolling and plastic deformation. These

[1] "Helical and Herringbone Speed Reducer Ratings," by W. P. Schmitter, presented at the annual meeting of the American Gear Manufacturers association, May 3-5, 1934.

INTENSE interest in the development and use of helical and herringbone gearing brings to significant prominence the suggested horsepower formula developed by the author of this article. His findings, though controversial, are based on sound fundamental knowledge backed by the research work of his company, The Falk Corp., and by years of practical experience in the field of gear design and application.

Recently Mr. Schmitter presented a paper "Helical and Herringbone Speed Reducer Ratings" before the American Gear Manufacturers' association. The accompanying article and its sequal, to be presented in the July issue, include some of the data incorporated in that paper, revised and supplemented to direct the contribution specifically to readers of MACHINE DESIGN.

Mr. Schmitter would be the last man to claim that his work represents the final word on helical and herringbone gears. His results, however, are the most up-to-date and conclusive that have been propounded. Comments on the analysis from leading authorities on gearing appear on pages 61 and 62.

—The Editors

will be discussed in detail later in this article.

It is obviously impossible to devise a design formula which is rational as regards both surface resistance and beam strength. Tooth breakage occurs but rarely due to the strength of helical gears; therefore, it is logical procedure to base initial design on an allowable surface pressure for wear. The pitch should then be selected by a comparatively simple design rule. Where a careful and detailed check of bending stress is warranted, one of the modified forms of the Lewis equation may be used, although not even these are rational for helical gears.

Considers Critical Point of Loading

At one time it was thought that the wear on a gear was directly proportional to the load put upon it and that the size was a direct indication of the number of years service which would be rendered. In 1920 J. Jandasek pointed out in a paper entitled "Gear Tooth Sizes From Standpoint of Durability," *Automotive Industries,* v. 42, pages 1305-9, 1402-6, 1412, that there apparently was a critical value of surface pressure for any given hardness, toughness and friction coefficient above which wear was rapid and below which the gear wear was negligible. This idea was developed further in his articles of 1921 and 1922 in *Automotive Industries* and the *S.A.E. Journal.* Further proof of such a critical point of loading was obtained from the rather complete series of tests on spur gears carried out by Ham and Huckert, and covered in University of Illinois Bulletin No. 149 entitled "An Investigation of the Efficiency and Durability of Spur Gears." Under conclusions, page 56, item 72 they state, "Apparently for any pair of gears there is a critical surface pressure governed by the properties of the materials above which the life of the gears is short and below which the gears will run indefinitely without appreciable wear."

If a method is available for ascertaining accurately just where the critical point of loading lies, and methods of control and manufacture are rigid enough to eliminate the possibility of falling short of certain minimum standards of manufacture, then a reasonable factor of safety is all that is required.

In 1928 the Falk Corp. completed a series of tests on herringbone gears in speed reducers. This work led to the unmistakable conclusion that a critical point of loading existed in helical gears, as well as in spur gears. The relation between wear and pressure is shown in *Fig.* 1. This point of loading is reached when the critical stress reaches the yield point or endurance limit of the material. Complete breakdown does not take place suddenly because of the difference in the strength of the individual crystals.

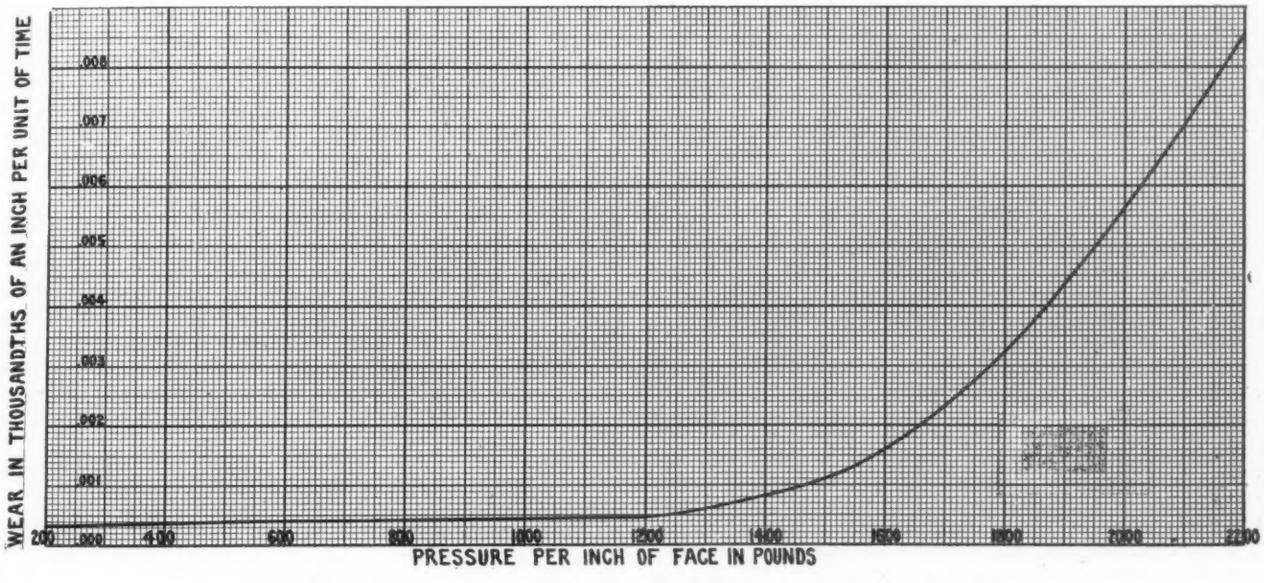


Fig. 1—*Wear data obtained from load tests indicates critical point of loading*

A cursory examination of the literature relating to the theory of contact stresses in elastic cylinders is sure to confound one due to apparent inconsistencies. The basic equation for the maximum compressive stress for two elastic cylinders, *Fig.* 2*A*, was developed by Hertz and is given by the equation.

$$S_{max} = 0.5825 \sqrt{P' \times (1/r_1 + 1/r_2) \div (1/E_1 + 1/E_2)}$$

where

E_1, E_2 = moduli of elasticity of the respective cylinders
r_1, r_2 = radii of curvature of the respective cylinders
P' = the compressive force per unit of length
S_{max} = the maximum specific compressive stress
Poisson's ratio = 0.25

Since the material at the center of the surface of contact is prevented from lateral expansion, it

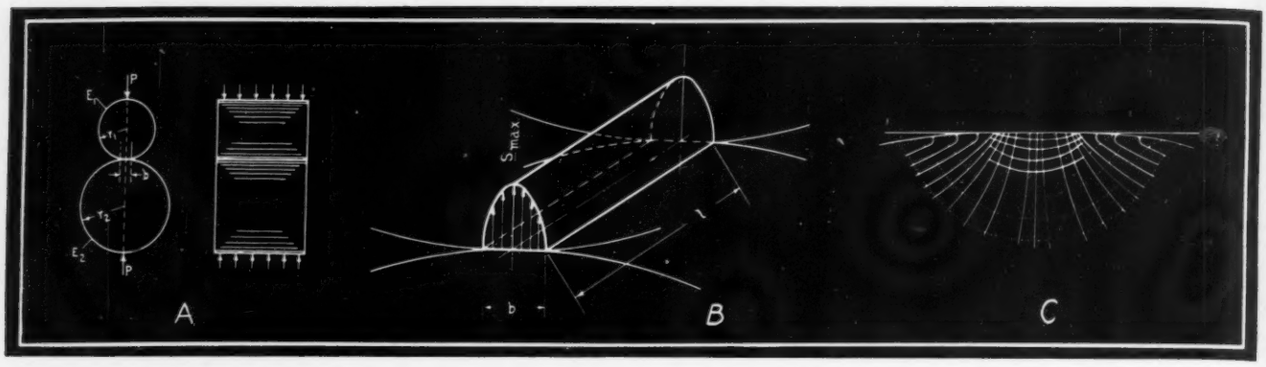


Fig. 2—A—Two elastic cylinders flatten at point of contact when subjected to compressive force. B—Variation and magnitude of maximum compressive stress may be represented by a semi-elliptical cylinder. C—Lines of principal stress at contact point of a compressed body show tensile and compressive stress components

is in a condition of compression from all sides and will sustain high pressures. Failure is therefore not the result of the direct compressive stress S_{max}.

In *Fig. 2B* is shown a semi-elliptical cylinder representing the stress magnitude at the various portions of the flattened area of contact b. The area of the ellipse being equal to π times the product of the semi-axes, the area of the stress diagram will be equal to

$$1/2 \times (\pi \times b/2 \times S_{max})$$

The total pressure is

$$P = 1/2(\pi \times b/2 \times S_{max} \times l)$$
$$= \pi/4 \times S_{max} \times b, \text{ when } l = 1 \text{ inch and}$$
$$S_{max} = 4P/\pi b = 1.272P/b$$

The width of the flattened area b can be obtained from the well known Hertz equation

$$b = 2.185 \sqrt{P' \times (1/E_1 + 1/E_2) \div (1/r_1 + 1/r_2)}$$

where Poisson's ratio equals 0.25.

Fig. 2C shows the lines of principal stress in the compressed cylinder. The full curved lines represent compressive stresses and the dotted lines are lines of principal stress in tension. Near the compressed area, both the principal stresses are compression.

It is well known that a body subjected to pressures acting at right angles to each other will fail due to shear when one-half the algebraic difference between the maximum and minimum stress components exceeds the yield point of the material, shown graphically in *Fig. 3A*.

Induced shear stresses in the compressed elastic cylinder are present as shown in *Fig. 3B*, both the intensity and the depth of maximum shear stress changing with the loading. The relations existing between the compressive, tangential and shear stresses have been investigated by Baud[2], Thomas and Hoersch[3], Belajef[4] and others. These physicists are not entirely in agreement. Belajef considers that a tensile stress within the body is numerically equal to the difference in stress components and that this is the critical stress. Baud states that the critical stress is shear and for spur gears is equal to $S_{max}/2$. However, all of his recorded results as obtained in the photoelastic tests show the maximum shearing stress to be approximately equal to one-half the difference of the principal stress components. It is possible that Baud has assumed this higher stress to take place at the extremities of the face or at the point of disengagement. The curling over at the edges of gear teeth long before there is any evidence of the usual types of surface distress would indicate that the stresses are higher at those points, although relief is obtained as soon as the edge gives way. In helical gearing especially, such a stress need not be considered because the load is simultaneously carried by all portions of the tooth.

An excellent treatment of the problem is that due to Thomas and Hoersch. They have investigated the subject completely from the mathematical angle and corroborated their results in the laboratory by the strain-etch method to within 2 per cent. It is significant that Baud's photoelastic results come very close to those obtained by

[2] "Contact Stresses in Gears," by R. V. Baud, *Mechanical Engineering*, Vol. 53, No. 9, pp. 667-674.

[3] "Stresses Due to the Pressure of One Elastic Solid Upon Another," by H. R. Thomas and V. A. Hoersch, University of Illinois, Bulletin No. 212.

[4] Prof. N. M. Belajef, Institute of Ways of Communications, Leningrad, Russia.

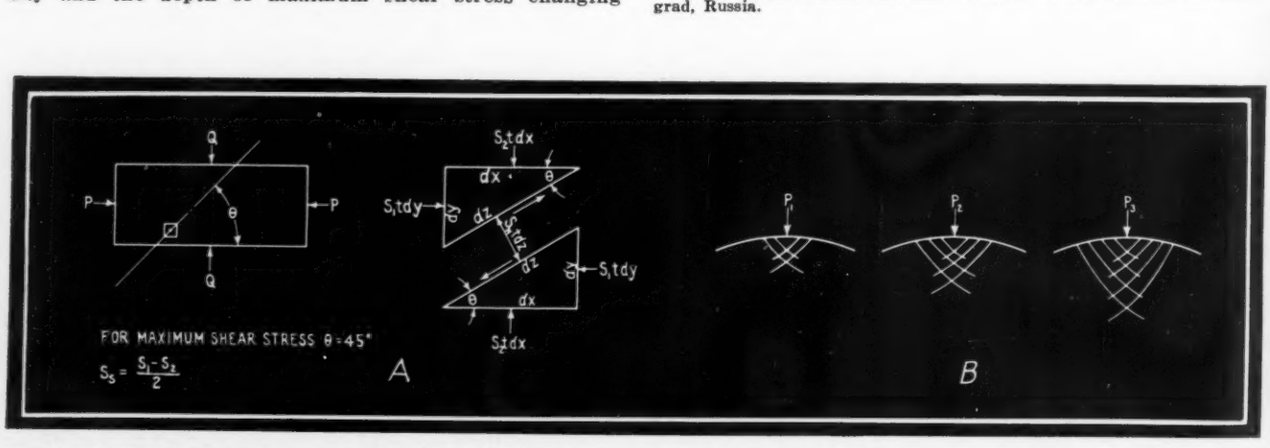


Fig. 3—A—Shear stresses are set up in a body when subjected to biaxial loading. B—Intensity and depth of maximum shear stress varies with load

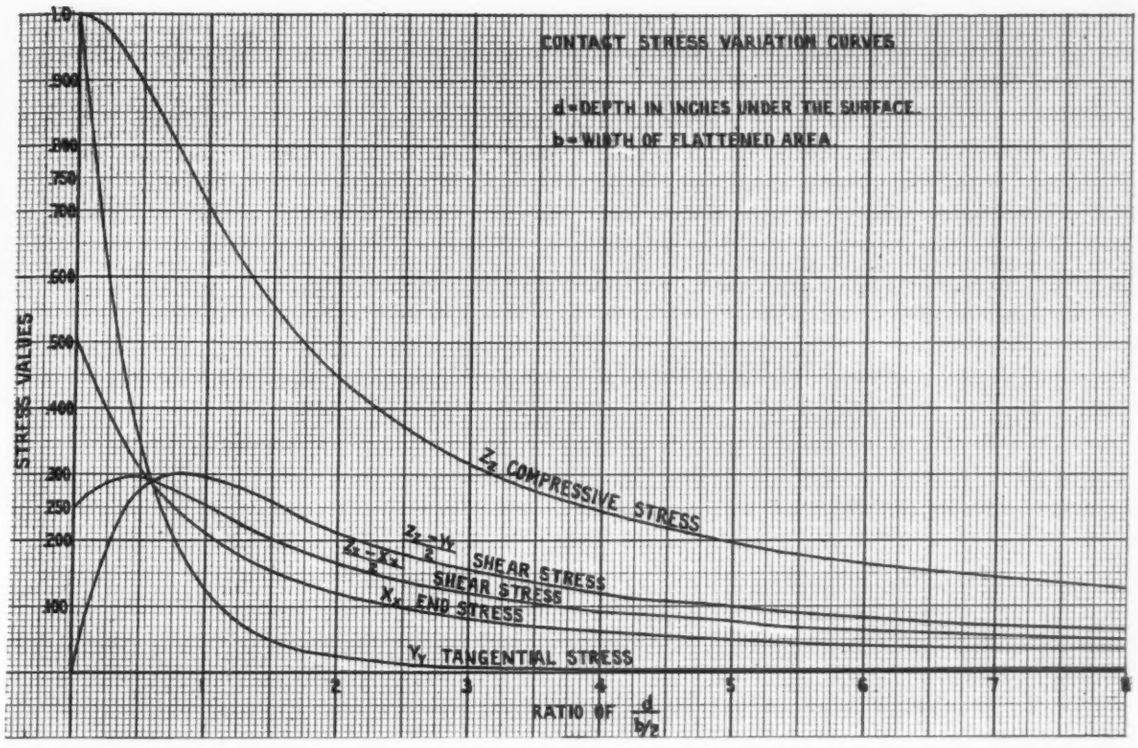

CONTACT STRESS VARIATION CURVES
d = DEPTH IN INCHES UNDER THE SURFACE
b = WIDTH OF FLATTENED AREA
STRESS VALUES
Z₂ COMPRESSIVE STRESS
Z₂ − X₂ / 2 SHEAR STRESS
Z₂ − Y₂ / 2 SHEAR STRESS
X₂ END STRESS
Y₂ TANGENTIAL STRESS
RATIO OF d / b/2

Fig. 4—Magnitude of stress components varies with ratio of depth of penetration to width of flattened area

Thomas and Hoersch. As a matter of fact, the conformity with the Thomas and Hoersch mathematical treatment is closer than with his own due probably to the fact that Baud has used a stress parabola rather than a stress ellipse.

Fig. 4 shows the relation of the various stresses in the elastic cylinder. The maximum shearing stress is obtained at a point 0.393*b* below the surface. The fundamental mode of stress variation is exactly the same for the various investigators, the question being largely as to the magnitude of the stress and whether it is compression, tension, or shear.

If we assume for the moment that the governing stress is compression, then some of the materials having a low modulus of resilience, such as cast iron, will indicate greater load carrying capacity than steel. Application of the various stress theories to speed reducers of four leading manufacturers indicates:

That if the governing stress is compression, the loads carried are approximately three times greater than they should be

That if the stress is tension, according to Belajef, the loads now being carried are more than 75 per cent greater than they should be

That if the governing stress is shear equal to ½ S_{max} the loads are 2.75 times greater than they should be

That if the governing stress is shear equal to one-half the difference of the principal stress components the loads closely agree with present practice.

There are one or two stress theories that make allowance for the effect of the normal pressure and the coefficient of friction on the magnitude of the tangential stress. The reduced stress of Fromm[5] is an example.

[5] "Zulassige Belastung von Reibungsgetrieben mit zylindrischen oder kegligen Radern," "Allowable Stresses of Friction Drives with Cylindrical and Conical Wheels," by Dr. Ing. Hans Fromm, Berlin.

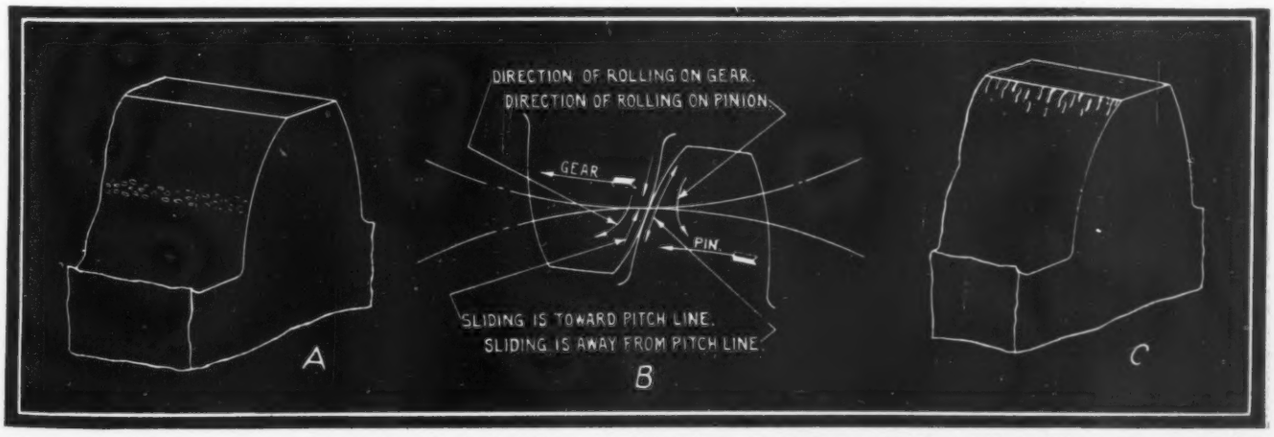


Fig. 5—A—Pitting of gear tooth at theoretical pitch line. B—Direction of sliding on mating tooth profiles reverses at pitch line. C—Scuffing takes place at extremities of gear teeth

Because of the difficulty in determining friction constants, it is considered that such refinements are impractical at this time. The velocity factor broadly circumvents the need for separate consideration of the various dynamic effects.

The development of the equation which follows is based on the theory that the critical stress is shear, and is equal to one-half the difference of the stress components. Perhaps the question is an academic one since in the final analysis the application of judgment and experience to the working factors could be so handled that any of the stress theories could be made applicable.

Thomas and Hoersch have shown that the maximum difference of stress components for an elastic cylinder in contact with a plane is

$$ {}_1Z_z - {}_1Y_y = -0.30025 \times b/\wedge $$

where ${}_1Z_z$ is the stress component below the surface directed toward the interior of the body and ${}_1Y_y$ is the stress component below the surface parallel to the contact plane. $\wedge$ is a term which takes account of the elastic properties of the two bodies and their shapes at the point of contact. The value for $\wedge$ modified for cylinders is

$$ \wedge = [(1-v_1^2)/E_1 + (1-v_2^2)/E_2] \times [2r_1r_2/(r_1+r_2)] $$

Where v_1 and v_2 = Poisson's ratio for the two cylinders. For two steel cylinders in contact

$$ \wedge = 4r_1r_2/(r_1+r_2) \times (1-v^2)/E $$

Since the maximum shearing stress is equal to one-half the algebraic difference of the stress components,

$$ S_s = 1/2({}_1Z_z - {}_1Y_y) = -0.150125 \times b/\wedge $$

Substituting the values of b and $\wedge$ for two cylinders in contact

$$ S_s = 0.150125 \left[\frac{2.185\sqrt{P' \times (1/E_1 + 1/E_2) \div (1/r_1 + 1/r_2)}}{\left(\frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2}\right)\frac{2r_1r_2}{r_1+r_2}} \right] $$

For two steel cylinders ($v = 0.25$)

$$ S_s = 958\sqrt{P' \times (r_1+r_2) \div 2r_1r_2} $$

which is the equation for the magnitude of maximum shearing stresses and which is otained at a distance $0.393b$ below the sufface.

At the surface the stress components are higher than those in the interior, but the stress differences are less. With Poisson's ratio equal to 0.25 the surface stress component ${}_oX_x$ is equal to one-half of the compressive stress component ${}_oZ_z$. The maximum shear stress at the surface is therefore

$$ S_{s\,s} = ({}_oZ_z - 0.5\,{}_oZ_z)/2 = 0.25\,{}_oZ_z $$

Surface and Sub-Surface Failures

Pitting. The surface of the tooth of *Fig.* 5*A* shows pitting of the type frequently encountered in helical gear practice. The exact cause of pitting is not definitely known, but its nature indicates that it is a shear failure. Pitting may be of a corrective or of a progressive nature. The former is experienced on gears even when they are loaded within their normal capacity. It should never be a cause of concern. Progressive pitting, on the other hand, is an extremely serious matter, and if it is not checked will soon render the gear incapable of performing its duty.

Lubrication at the pitch line is poor due to the fact that the oil wedge is broken at that point by the change in direction of sliding, *Fig.* 5*B*. The duration of contact is at the same time greatest at the pitch line. It is quite likely that the material at the point of greatest shear stress is worked harder than at other profile portions.

There is some question as to whether a stress within the tooth ($0.393b$ below the surface) greater than the endurance limit of the material, can start fatigue cracks if the stress at the surface is less than the endurance limit. Since it is on the side of safe practice, maximum shear stress below the surface had best be used as a basis of loading.

Spalling. In the case of great overloads the maximum stress occurs at a greater depth, and as a consequence the size of the dislodged particles are much larger than in pitting. The distress is not limited to the vicinity of the pitch line, but extends over the entire tooth.

Wear. There are three general types of wear; physical, chemical and abrasive. The first probably takes place when the shear stress at the surface exceeds the strength of the crystals, the second is due to recurrent formation and removal of oxides and sulphides, and the third is caused by scratching of foreign matter such as silicates. With proper lubrication abrasive and chemical wear are never serious.

Scuffing — When the pressure velocity values become excessive, especially where lubrication is inadequate, a drawing action at the extremities of the teeth is experienced, *Fig.* 5*C*. This may occur in either the recess or approach, and is sometimes found in combination with pitting.

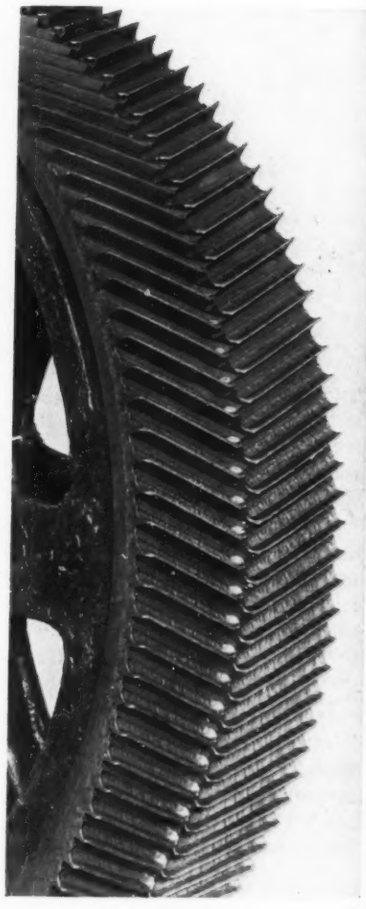


Fig. 6—Typical failure of gear teeth caused by rolling

Scuffing is caused by a combination of compressive and tensile surface stresses. Before it can take place there must be an adhesion of the surfaces of the mating profiles. The tendency of the incoming tip of the driven tooth to scrape the oil from the sides of the driving tooth is an important contributing factor and calls for tip relief. Smooth finish will aid in preventing the occurrence of scuffing.

The stresses responsible for scuffing bear sufficient relation to the shear stress, so that if the pressure velocity value is controlled with the aid of a satisfactory speed factor, distress of this type will be absent.

Rolling. In rolling the system of forces producing the alteration is similar to that present in scuffing except that complete failure of the lubricant has taken place with subsequent seizure of the mating profiles. In rolling there is a complete rearrangement of material as can be seen from *Fig.* 6.

Plastic Deformation. Under excessive momentary overload the profile of a soft gear may be concaved. The possibility of this type of failure makes it necessary to limit the shear stresses resulting from the maximum load to a value less than the shear yield point of the material.

Gouging. With insufficient tooth correction, and under too high a sustained pressure, the sharp edge of the entering tooth of the driven member digs into the tooth of the driver at a point just below the pitch line. A similar action is sometimes responsible for gouging on the driven tooth in the arc of recess. Extreme pitting frequently is found in combination. This trouble is not incurred if loads are kept within normal values.

Load Distribution

Since the gear and pinion teeth are carrying load in different portions of the contact zone, the deflection of the tooth as a beam will to some extent influence the portion of the load carried at any particular point on the surface. There is a slight tendency to load up the center portions of the teeth because the deflection of the beam

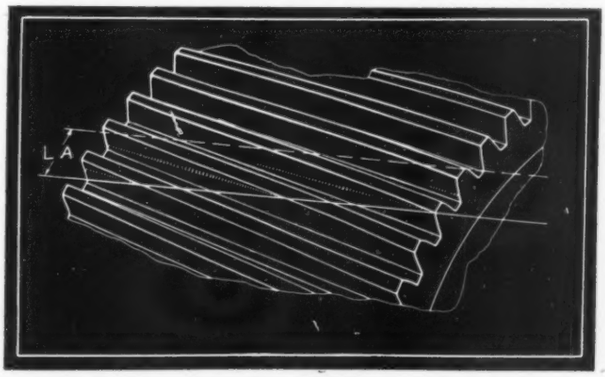


Fig. 7—Tooth deflection results in slight variation of load distribution with heaviest at the pitch line (exaggerated)

is somewhat less when the teeth engage at their middles. This is shown diagrammatically in *Fig.* 7, where the width of the contact strip is shown the greatest in the center of the tooth. It would appear to be an exceedingly difficult matter to evaluate this load distribution. Experience indicates that the decrement of loading toward the root and tip is quite small so its exact determination is neglected. Due allowance for these small stress differences is made in the inbuilt factor discussed further on in this article. It would be expected that longer teeth would show greater difference in loading, but this is not entirely proved out in practice. Apparently the thin, long tooth yields more readily and at least partially compensates for any expected difference in load distribution.

Dynamic and Static Stress Relations

It has long been known that dynamic loads on gear teeth are greater than what they would be if the gears were only statically loaded. The work of Professor Buckingham and the A.S.M.E. special research committee on strength of gear teeth is outstanding in this connection in that it is the first time that the dynamic loads have been evaluated. In applying the results of Buckingham's work to spur gears, the load is first computed based on static pressure, and to this is added the dynamic load as computed from constants developed in this test work.

The constants of rigidity are as important a factor in the determination of the magnitude of the dynamic stresses as are the tooth errors. It is possible to assume the extent of the spacing errors on the gears which are being designed because they will be made on a particular type of gear cutting machine with cutters or hobs of a known standard, and with manufacturing control assured by the necessity of meeting inspection requirements. The determination of the elasticity of the system and the effective masses is quite another matter however, since they can be obtained only by extremely laborious computations to say nothing of the difficulty of ascertaining the physical dimensions upon which to base them. For this reason computations of dynamic loads of spur gears are based on an *assumed* average condition for the elastic system.

In the herringbone field there prevails a flexible standard of accuracy dependent on the speed. Refinements in the case of gears operating at 5000 to 10,000 feet per minute call for the ultimate in accuracy of gear cutting on machines reserved for this class of work. The manufacturing department dares not ignore the specified tolerances of even a single set of gears because they will only be rejected by the test department due to noisy operation. The practical aspect of gear accuracy requirements has centered itself about what is acceptable as regards noise.

Well Known Hobbers Used

In the helical and herringbone fields gears are cut on a few well known makes of hobbers and shapers so that variations are even less than those which obtain in the spur gear field. It would appear practical therefore to assume a fairly rigid standard of accuracy among the various manufacturers for the various speed ranges to which their product is applied. Smoothness of operation with helicals is not as much a matter of tooth profile as in the case of spurs because of the axial overlaps and thus the dynamic loads are proportionately lower because of the soft nature of the engagement.

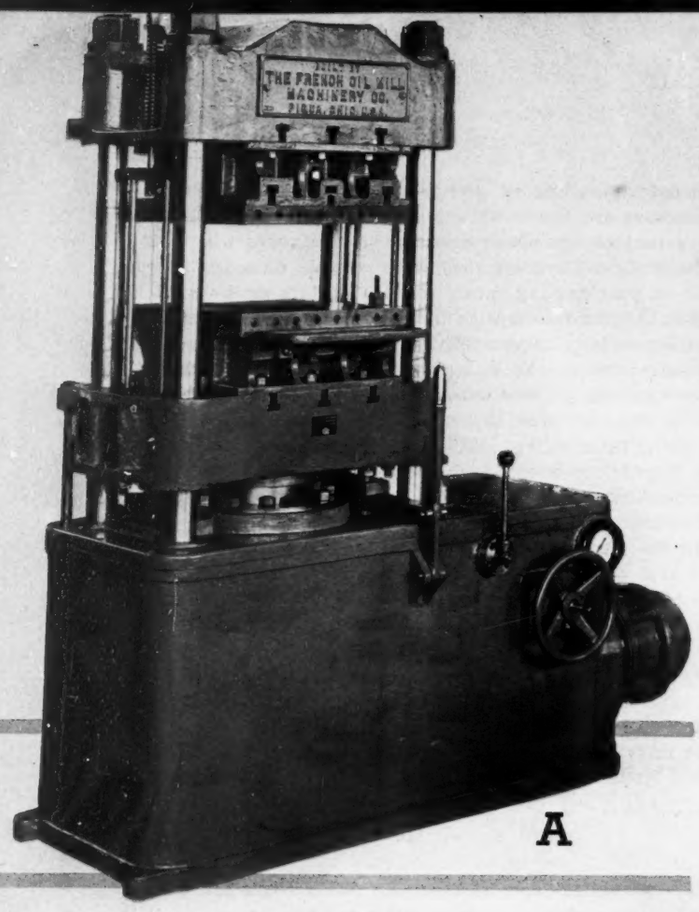
If these are all fair assumptions then there is sufficient license to avoid separate computation of the dynamic loads, providing it is possible to find a modifying factor which will correctly express the relation between static and dynamic loads for the prevailing standards of accuracy at various speeds, and for assumed average conditions of rigidity of the elastic systems. This factor cannot be determined from measureable values because unfortunately no such experimental data exists for helical gears as has been given us for spur gears. There is however a wealth of experience upon which to draw. A comparison of the general practice as regards loading for various speed ranges will indicate the degree of derating which has been found necessary to take care of changing dynamic conditions.

Careful examination of the various velocity factors indicates that one of them has a wide range of application, and grees with prevailing practice quite closely. This is the rule C_v, the velocity factor, equals $78/(78+\sqrt{V})$, where V is the pitch line velocity in feet per minute.

One factor alone is not sufficient to take care of the relation of dynamic to static stresses. An additional factor, C_i, the combined inbuilt factor, makes allowance for the fact that the tooth errors increase with an increase in the size of the gears. It also provides for other practical deviations which of necessity limit any theoretical design equation.

It is increasingly difficult to obtain optimum contact if the physical dimensions of the gears are increased. The mounting is as a rule more accurate on a small set of

(Continued on Page 53)



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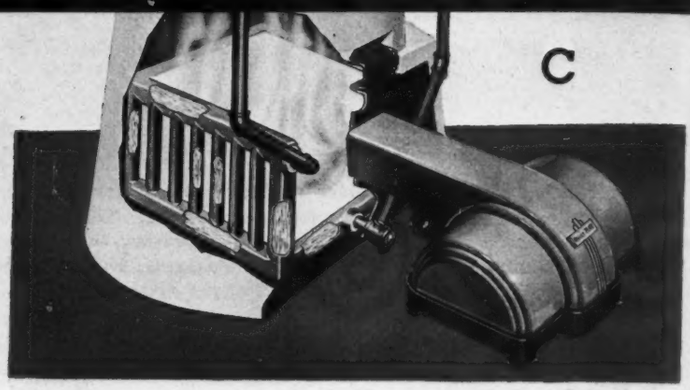
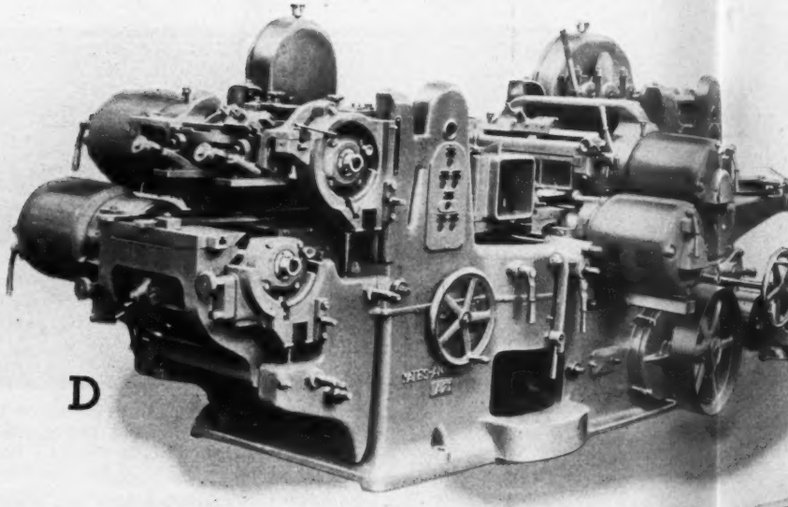


C

Design Features in New Machines

A Pictorial Presentation of Recent Machinery from the Standpoint of Design.



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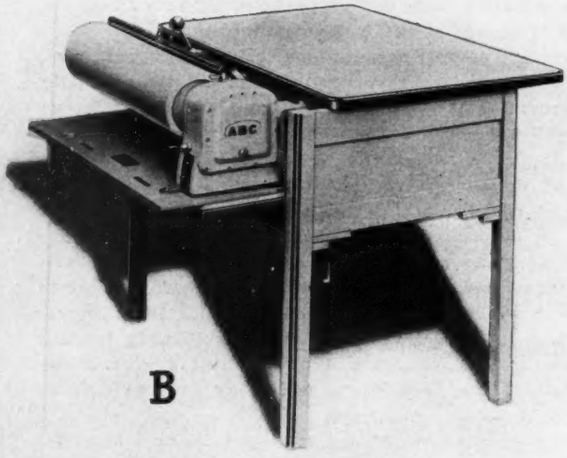

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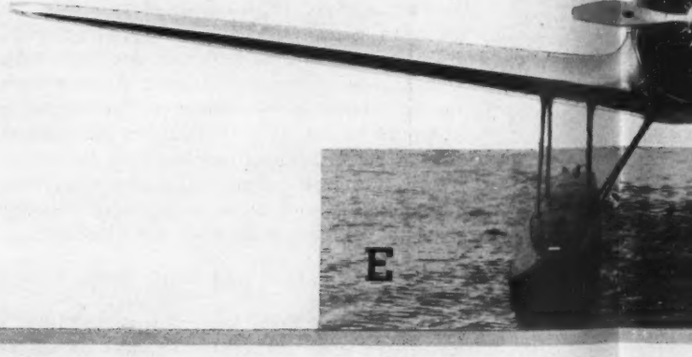


E

(A) All valves, the cylinder and pump of plastic molding press built by French Oil Mill Machinery Co. are enclosed in the cast iron base which also serves as a reservoir for the oil employed in hydraulic operation. An automatic pressure regulator eliminates safety and relief valves. The machine is used in the molding of phenolic plastic products.

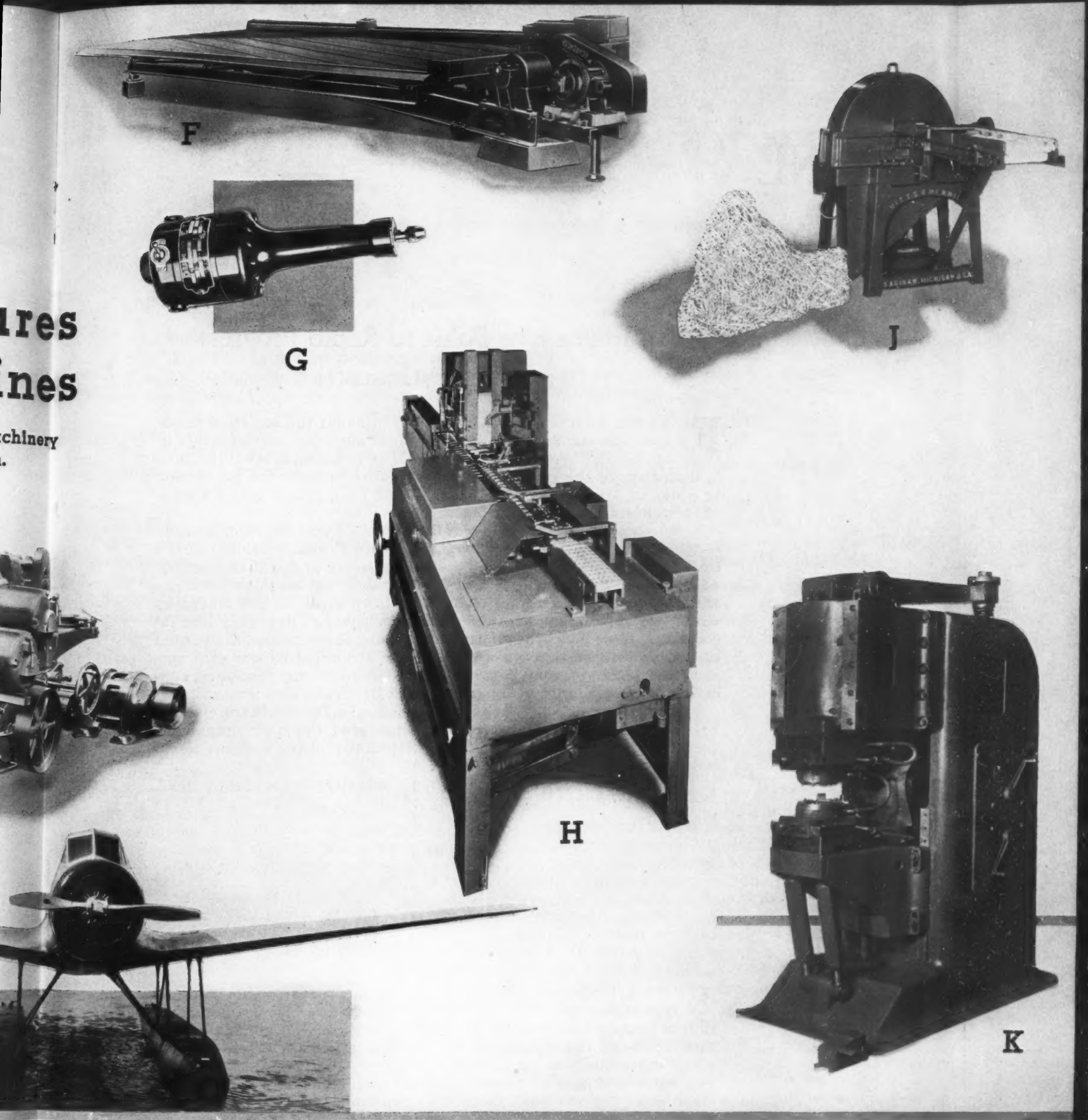
(B) Parts used for engaging and disengaging movements in ironer of Altorfer Bros. Co. are of high grade manganese steel. Polished, chromium plated ironing shoe is hardened. The head is of die cast aluminum. All moving parts run in self-lubricating bushings. Controls are placed to permit convenient operation.

(C) Compactness without sacrifice of accessibility is achieved in the design of the oil burner introduced by May Oil Burner Corp. The pump is driven by a split phase motor through a shaft articulated at three points. In this way unequal thrusts are eliminated.

(D) Feed works of new planer and matcher consists of a combination of chains, sprockets and gears with shafts and sprockets rotating on roller bearings. Chipbrakers are spring loaded and have quick release. The countershaft is carried in the infeed table unit which is bolted to the machine base at the top only, thus eliminating transmission of vibration to the machine, made by Yates-American Machine Co.

(E) Wing flaps are employe landing speed of the ne Aviation Mfg. Corp. This pl low-wing, all-metal type, e pitch propeller. Landing gear tion may be adjusted by each

(F) Fewest possible number adjustment are incorpora of the sand and gravel washin Concentrator Co. In this m translated into a differential re the strokes being at the same of the head motion. Main b while the crank is lubricated



F

G

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K

s are employed to bring down the peed of the new seaplane of General Corp. This plane, of the cantilever, metal type, employs a controllable Landing gear is retractable. Ventila- usted by each passenger.

sible number of parts and only one are incorporated in the head motion gravel washing table built by Deister . In this mechanism, rotation is differential reciprocating movement, at the same rate as the revolutions ion. Main bearings are chain oiled is lubricated by a patented impulse pump. A short center multiple V-belt drive transmits the power from the motor.

(G) Dynamically balanced armature of the motor used in this hand grinder is mounted in double grease sealed ball bearings. The universal motor is ventilated so that continuous use is possible. Made by Dumore Co., the grinder is balanced carefully.

(H) Cartons move continuously in one conveyor and their intended contents move continuously and synchronously in a parallel conveyor in the loading cartoner of F. B. Redington Co. Contents are gradually transferred to the carton as both conveyors travel at a fixed rate of speed.

(J) Width of excelsior cut by paper shredder of Mitts & Merrill is adjusted by a variable speed mechanism which regulates the speed of the feed conveyor. Use of a high torque, low starting current motor permits the specification of a small motor for starting the knife disk. The motor is mounted on a floating base.

(K) Control of the new heavy duty, pneumatically operated press welder of Taylor-Winfield Corp. is fully automatic, the operator merely inserting the pieces to be welded, tripping a foot switch and removing the complete welded article, each function of the welding operation being performed in proper sequence.

MACHINE DESIGN

L. E. JERMY, EDITOR
ALLEN F. CLARK
HAROLD B. VEITH
F. H. BURGESS

All Indications Point to Rapid Progress in Machinery Industries

FOR some months little has been heard about the so-called retrogression in standards of living brought about by developments in machinery. The tide has turned. All evidence on this subject now is indicative of the need for increased effort to facilitate production in order to bring everyday requisites and even luxury items within the purchasing power of the majority.

Outstandingly illustrative of this new—or perhaps "considered"—thought is a statement made by Dr. Glenn Frank, president of the University of Wisconsin, at the opening meeting of the 1934 Century of Progress: "The machine has not betrayed us; we have betrayed the machine! Science and technology have given us the means by which we may emancipate the race from poverty, drudgery and insecurity. If we now prove incapable of using these means, the verdict of history will be that we were a people strangled by our own success." The 300 present, comprising leaders in industry, science, education and medicine, rose up and cheered Dr. Frank's statements.

Changes in policies of the government also illustrate the change-over. No longer is the purchaser considered the only influencing factor in recovery. Manufacturers, especially in the durable goods industries, are in the eye—to stay.

Technocracy, as understood when originally propounded, is dead. We are in another phase: Recovery of country and of the machinery industries.

• • •

Watch Gearing Developments!

FEW machines are designed that do not incorporate gearing of one type or another. It is not surprising therefore that developments in this form of transmission recur constantly, or that gears as a subject command the interest of designers to a greater extent than do many other machine parts.

Increased use of helical and herringbone gears, particularly since attention has been focused on attempts to design quieter and more efficient machines, brings to the forefront the accomplishments of one of the leading engineers on this particular type of gear. Based on the results of his work, which he discussed at the recent meeting of the American Gear Manufacturers' association, this authority has covered the subject of helical and herringbone gearing especially for MACHINE DESIGN.

The article commences on page 48. Though a composite whole and difficult to break into sections, the contribution necessarily is being published in two parts on account of its length. The second section will appear in July.

PROFESSIONAL VIEWPOINTS

Machine Design Welcomes Letters Suitable for Publication

Design Features in New Machines

To the Editor:

CONGRATULATIONS on bringing together a number of photographs showing interesting new machines and on your method of presenting these as the center spread in MACHINE DESIGN!

I find that we still have a long way to go toward the end of producing good appearing machines. It is a recognized fact that the appearance of a machine is one of its most important sales points. Still, a study of your feature page shows that great strides in this respect have yet to be made. Machine designers may well profit from this study.

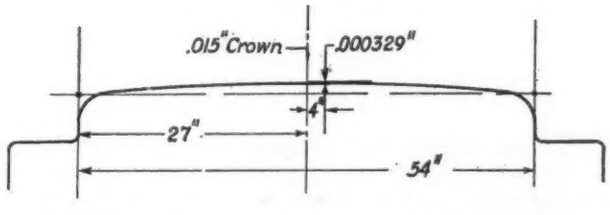
May I suggest that you do not "bleed" the photographs so severely, since by so doing you take some of the value away from these illustrations?

—ARTHUR J. STOCK,
Stock Engineering Co.

Simplifying Design Calculations

To the Editor:

THE finding of drops in a shallow curve by the ordinary method of triangles is a rather complicated procedure, since the radius is extremely large. These drops can be determined, however, simply and with fair accuracy by a short cut method in which other designers may be interested.



Simplified computations permit easy determination of drop in curves of any radius

The steps in the method are as follows: Multiply the squared distance of the required drop from the center line of the curve by a constant equal to the height of the curve and divide by the squared half-chord.

As a practical illustration, let it be required to find the drop four inches from the center line of a crowned roll the length of which is 54 inches and the crown of which is 0.015 inch. The radius corresponding to this curve would be 2430.0075 inches, which would evidently lead to long and tedious figuring, whereas by applying our formula we have:

$$4^2 \times \frac{0.015}{27^2} = 16 \times \frac{0.015}{729} = 0.000329 \text{ inch}$$

The long-hand method of calculation would give us 0.000328 inch.

The result obtained by aid of the formula is therefore sufficiently accurate, and besides it lends itself to a slide rule operation if we limit the answer to two or three significant figures.

—B. W. BARISH,
E. G. Budd Mfg. Co.

Spring Errors Eliminated

ELIMINATION of inherent errors encountered in precision springs is accomplished by the development of a new type of material by John Chatillon & Sons, New York. The errors eliminated are the temperature error, the straight line error, creep effect and hysteresis, and back error. Elinvar is the basis for the new material with the final development containing less chromium and additions of several elements including silicon and molybdenum to provide greater precipitation hardness.

The springs are made from flat wire of rectangular cross section, having the major and minor dimensions properly proportioned, each in respect to the other and both in respect to the diameter of the coil. Excellent elastic properties are obtained by cold working and after-heating.

"Contrary to general belief, more progress is made in efficiency during one year of depression than during five years of great business activity. The firm that emerges from one of these adjustment periods with more efficient and more economical methods, and with new and better products, will be the one most likely to succeed in the next era of prosperity which is sure to come. Preparation for future success takes the form of original research carried on assiduously."—P. S. Burgess, director, Arizona Agricultural Experiment Station.

TOPICS

SENSATIONAL accomplishments in centrifuge design are in the eye. One unit recently constructed from the design of Dr. Beams is said to run at 1,200,000 revolutions per minute and has a rim velocity of 1390 miles per hour. According to Arthur D. Little's *Industrial Bulletin,* centrifugal force is computed to be 7,600,000 times gravity, making a single ounce press against the walls of the machine with a force of 237 tons.

An amazing addition to high speed centrifuges is an attachment worked out by Harvey of Princeton and A. L. Loomis, whereby a microscope can be applied for observing the action of the centrifuge on a specific whirling object as though it were standing still.

* * *

In the new streamlined street cars built for Chicago the motorman will depress a foot pedal to obtain acceleration. A newly developed system of electric and hydraulic braking also will be incorporated in the cars. The motors are of an improved lightweight high-speed type, spring suspended in rubber-insulated truck frames and geared to the axles in silent, oil-filled housings.

* * *

There is little or no prospect of beryllium becoming a useful addition to the group of metals used in the manufacture of special steels, according to a test report on the effect of the addition of beryllium to steel. The investigation was made in England through the co-operation of the research departments of English Steel Corp. Ltd., Thomas Firth and John Brown Ltd. Metallic beryllium of 98 per cent purity was supplied by the Air Ministry which solicited the tests.

Beryllium is expensive and is unlikely ever to become reasonably cheap, while its lightness and readiness to oxidize made the actual introduction into molten steel somewhat difficult and distinctly wasteful, the report brought out. In regard to forging, addition of 1 per cent of beryllium tends to make some of the steels, particularly the 3½ per cent nickel and nickel chromium steels, distinctly "stiffer" under the hammer with an increased tendency to "burst" toward the top end. With any treatment none of the steel containing beryllium showed the necessary combination of high elastic limit and fatigue resistance with ductility and toughness.

* * *

When he awoke from a dream he had recently about his machine, Charles V. Boys, British scientist, hastened to his laboratory and incorporated the idea that came to him in his sleep. A foolproof device for measuring public consumption of gas for light and power was produced as a result.

* * *

Stainless steel is finding increased use in the domestic appliance field. One application is for the cooling units of electric refrigerators. A type designed and adopted as standard by an important builder utilizes 18-8 chrome nickel sheets, formed and fabricated by resistance welding.

* * *

Considerable interest has been manifested in a machine for visual demonstration of economic conditions. It has been designed by Dr. H. C. Dickinson of the bureau of standards. Claim is made that the "Economonstrator," as the device is called, will show when and why the economic system needs correction and also where and how correction might be applied.

* * *

A recent dispatch relates that many farmers have not replaced implements in three years. An investigation in Kansas fixes $500 as the minimum average for implement replacements when the farmer has the money.

MEN OF MACHINES

FORTY-TWO years in the profession of a gear maker, John Christensen now becomes president of the American Gear Manufacturers association which he helped found in 1917. He plays a prominent role in the gear manufacturing industry as president of the Cincinnati Gear Co., Cincinnati.

Born in Denmark, Oct. 14, 1871, Mr. Christensen left home at the age of 14 to become a sailor. After five years he came to this country. In Philadelphia he obtained a job with the Bilgram Gear & Machine Works. Starting as an operator of a gear cutter, Mr. Christensen gained a wide experience in gear technology.

In 1908 Mr. Christensen located in Cincinnati to form the Cincinnati Gear Co. He has been an active and enthusiastic member of the gear association since its inception.

JOHN CHRISTENSEN

• • •

R. H. ANDEREGG

VERITABLY an encyclopedia on data pertaining to heating, pumping and air conditioning—that's Reuben H. Anderegg. This enviable knowledge has been gained through eighteen years with the Trane Co., LaCrosse, Wis. Recently he became this company's engineering vice president.

During his career with Trane Mr. Anderegg has divided his time between sales engineering and design work. His record embodies many years of experience in design of pumping, heating and air conditioning equipment. Numerous articles on condensation and vacuum pumps have been contributed by him.

Technical society affiliations include the A.S.H.V.E. and A.S.M.E. About half of his time is spent on the road on large projects and in creating new fields for his company's products.

• • •

INDUSTRIAL designers, of whom Donald R. Dohner is one of the leaders, are contributing extensively to progress in design of machines. Mr. Dohner's success in this field was assured when years ago he set out to prepare himself to meet present day industrial and merchandising needs. In 1910 he left college to attend the Art Institute in Chicago. Not interested in "arty" art, he went to work in 1914 to apply his knowledge of design to everyday things. By working in woodworking and machine shops in Chicago and Indianapolis he acquired the practical aspect of engineering and production.

Ensuing years found him teaching at Carnegie Tech, acting as a consultant and later as director of art in the Westinghouse engineering department. This position he recently resigned to

D. R. DOHNER

set up an industrial design and research service with Alexander J. Kostellow.

Dohner insists that rational economic design starts prior to the drawing board and has its beginning in a thorough study of human needs, markets, economics, materials, tools and processes. He has applied this principle in the development of a wide range of machines.

* * *

R. E. COLE, one time chief engineer of Dodge, Durant and Rockne, has joined the Hupp engineering staff.

* * *

A. C. CHRISTENSEN has been made vice president and chief engineer of National Engineering Co., Philadelphia.

* * *

E. P. WELLINGTON, chief engineer of Rempe Fin Coil Co., Chicago, has resigned.

* * *

A. C. FULTON is vice president and chief engineer of the newly formed Ackerman Rubber & Plastic Molding Co., Cleveland.

* * *

RAYMOND LOEWY, industrial designer has been retained by Standard Oil Co. of New Jersey. He is body designer and consultant stylist for Hupp Motor Car Corp.

* * *

CLYDE E. WILLIAMS has been appointed director and Dr. H. W. Gillett as chief technical adviser of Battelle Memorial institute, Columbus, O.

* * *

W. H. BASSETT, American Brass Co., Waterbury, Co., Conn., has been nominated president of the American Society for Testing Materials.

* * *

DOUGLASS ANNIN, formerly chief engineer of Neilan Co. Ltd., Los Angeles, has joined the industrial regulator division of Minneapolis-Honeywell Regulator Co., Minneapolis.

* * *

R. B. MILDON recently was elected vice president in charge of the operations of marketing, engineering, manufacturing and service of Westinghouse, having to do with the products of the South Philadelphia Works.

* * *

ARTHUR HALSTED, for 14 years with the United States bureau of standards, has been appointed a member of the staff of American Standards association. He will have his headquarters in Washington.

* * *

GUNNAR E. PALMGREN has been appointed chief engineer of S. K. F. Industries Inc., Philadelphia. He comes from Sweden, where he has been connected with the company in a similar capacity. Mr. Palmgren succeeds H. E. Brunner, retired.

* * *

C. M. SAEGER JR., physicist, United States bureau of standards will prepare the annual exchange paper of the American Foundrymen's association to be presented before the annual meeting of the Institute of British Foundrymen, London.

* * *

DR. D. S. JACOBUS, advisory engineer, Babcock & Wilcox Co., New York, has been elected president of the American Welding society. J. J. CROWE, engineer in charge of research and development, Air Reduction Sales Co., Jersey City, N. J., is senior vice president.

* * *

JOHN W. O'LEARY, Chicago industrialist, has been re-elected president of the Machinery and Allied Products institute. Other officers re-elected are: H. C. BEAVER, Worthington Pump & Machinery Corp., vice president; PAUL C. DEWOLF, Brown & Sharpe Mfg. Co., secretary; and ROBERT H. MORSE, Fairbanks, Morse & Co., treasurer.

* * *

WILLIAM E. SYKES has been awarded the Edward Longstreth medal by Franklin institute in consideration of the design and development of the Sykes gear generator. He has been associated with Farrel-Birmingham Co. Inc., Buffalo, since 1922, when he came to the United States from England.

* * *

S. G. MCALLISTER has been elected director and first vice president of the International Harvester Co., Chicago. He has been with the company since 1897. From 1914 to 1932 he was assistant works manager in Europe, returning to become vice president in charge of manufacturing and engineering departments.

Obituaries

DR. h. c. DR.-ING. E. h. OSKAR VON MILLER, renowned engineer, died recently in Munich, Germany. He was an honorary member of the American Society of Mechanical Engineers and held the golden Grashof medal of Verein Deutscher Ingenieure. Dr. Miller was born in Munich, May 7, 1855. During a trip to this country in 1883 he met Edison and on his return to Germany he became technical manager of the Deutsche-Edison Gesellschaft in Berlin. After seven years in that capacity he returned to Munich to become a consulting engineer for the design and erection of electric power plants. Through his late years he was best known as the creator of the Deutsches Museum in Munich, which is made up of master works of natural science and engineering, illustrating their historical development.

* * *

Major William J. Hammer, distinguished engineer and one of the early associates of the late Thomas A. Edison, died recently. His war service included work at the United States patent office. While there he marked and held up certain patents likely to convey information to the enemy. An enthusiast in the science of aeronautics, Maj. Hammer became the owner of the first airplane ever sold in this country to an individual.

Determining Capacity of Gearing

(Continued from Page 45)

gears than on large ones. The difficulty of obtaining optimum physicals, increases with the size of the heat-treated gear. The face width has been used as the criterion for the C_f value because it apparently bears more importance in the determination than do the other physical dimensions.

For speed reducer work separate C_f factors are used, depending upon the position of the gear. The permissible loading is higher on the second reduction of a double reduction train, partly on the assumption that the dynamic stresses are greatest on the gears next to the relatively heavy driving masses. There seems to be some difference of opinion with regard to this question however. Careful analysis of a number of speed reducers made by various gear manufacturers indicates that the practice of loading the second reduction set higher than the first reduction set is practically universal. It is possible that some of this can be explained by the fact that there is a desire to play safe on the smaller and cheaper high speed gears. A little shorter life on the low speed gears in terms of number of tooth contacts is not open to serious criticism.

It should be remembered that the speed reducer is a precision machine, tooled and standardized, and with the mountings, lubrication, etc. under accurate control. *For general application*, the C_f curve should be used the same as for the initial set of speed reducer gearing, having a value of approximately two-thirds at the smallest face widths and 0.425 for face widths of 18 inches and larger.

The instantaneous radii of the contacting teeth determine the curvature of the elastic cylinders. Although the tooth shape is involute and the curvatures constantly changing, calculations should be based on the curvature at the pitch point because that is where failure first manifests itself. Since the contact extends diagonally up the tooth at an inclination *VBH*, *Fig.* 9, and is a straight line, it must be used as the length of the contacting cylinder. The curvature of the tooth must be determined in a plane at right angles to this contact line as shown in *Fig.* 9.

Fig. 8—Diagrammatic representation of the more important angular and pitch relationships in helical gears

In considering contact between involute spur gear teeth, the radius of curvature of each gear in a plane perpendicular to the line of contact is the distance along the normal to the tooth (line of action) from its point of intersection with the tooth profile to its point of tangency with the base circle. This is illustrated in *Fig.* 9. The radii of the tooth profiles at the pitch line for spur gears are equal to $(PD_p \times sinVP)/2$ and $(PD_g \times sinVP)/2$.

In helical involute gears the radius of curvature of each gear in the plane perpendicular to the line of contact is the distance along the normal from its point of tangency with the base cylinder to the tooth profile. The normal to the generating rack at any point of contact with the gear tooth will be tangent to the base cylinder. Thus the curvature at the pitch line of helical gears is

$$(PD_p \times \sin VP) / (2 \cos VBH)$$
$$\text{and } (PD_g \times \sin VP) / (2 \cos VBH)$$

In helical gears all phases of contact are simultaneously present as shown in *Fig.* 10. The total length of contact has an important function in determining what the load carrying capacity of the gear will be. This length is at one time a maximum and at another time a minimum. The highest unit loading is obtained when minimum contact is present.

The theoretical determination of contact variations is an important design problem. With inadequate consideration of the elements involved highly unsatisfactory results are apt to be obtained. *Fig.* 12 shows the ratio of maximum and minimum contact lengths plotted against face width. Peculiar shaped curves for 1, 5 and 10 D.P. show at the lower portion the

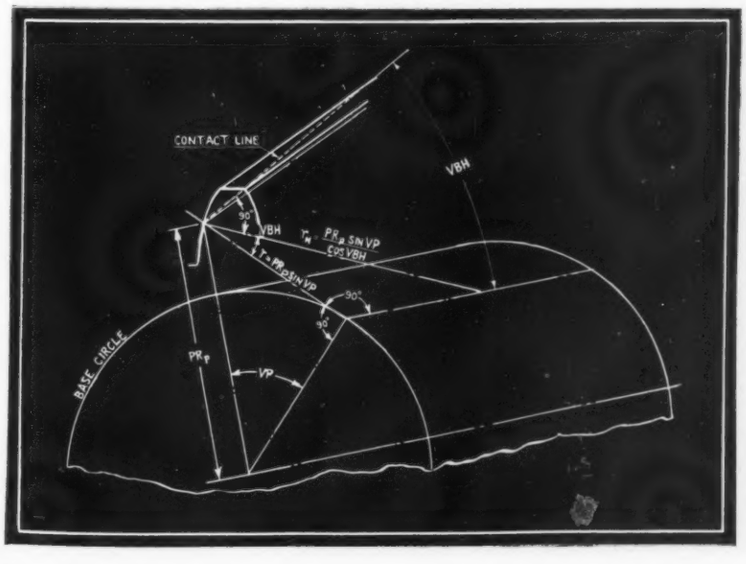


Fig. 9—The curvature of the helical tooth at the pitch point in a plane normal to the contact line is a function of the pressure angle and the base helix angle

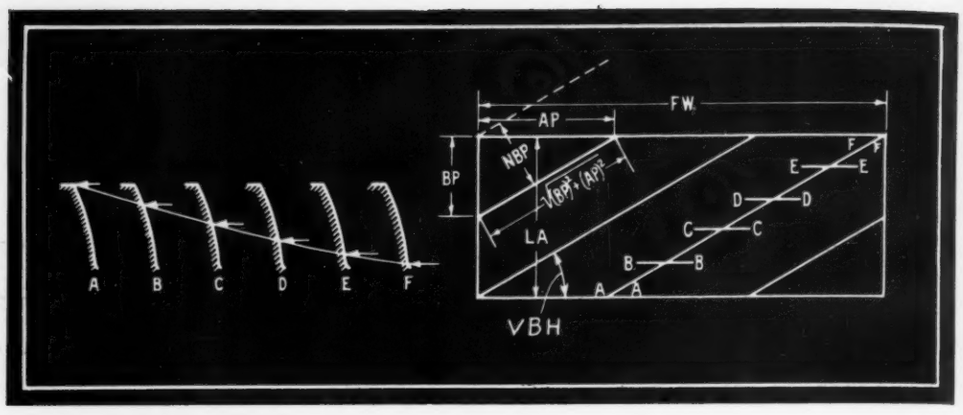


Fig. 10—All phases of contact are simultaneously present in helical gearing

ratio of minimum contact length of face plotted against face width, and at the upper portion the maximum contact length ratio obtained for any particular face width. It is apparent that when gears are improperly chosen, considerable variation can be obtained in the relation of minimum to maximum contact length. The result is that the capacity is less than what might ordinarily be supposed because of the low value for minimum contact. In addition, excess variation in contact often results in setting up stress variations over a greater range than need be.

The length of the line of action is found by the well known expression

$$LA=\left[\sqrt{(OR_p)^2-(BR_p)^2}-\sqrt{(PR_p)^2-(BR_p)^2}\right]+\left[\sqrt{(OR_g)^2-(BR_g)^2}-\sqrt{(PR_g)^2-(BR_g)^2}\right]$$

with symbols as indicated in *Fig.* 11.

The average number of teeth in contact in the plane of rotation is

$$N_r=LA/BP=LA/(CP\times\cos VP)$$

where

BP = base pitch
CP = circular pitch
VP = pressure angle

The contact zone can be represented by a rectangle, *Fig.* 10, having for its sides the line of action LA, and the face width FW. The teeth are shown passing through the rectangle at the inclination VBH, which represents the base helix angle. The distance between adjacent teeth in the plane of rotation along the line LA is equal to the base pitch BP. The normal spacing of the teeth is equal to the normal base pitch NBP. From the trigonometry of the figure, it is evident that

$$VBH=arc\ tan\ (BP/AP)$$

$$NBP=(BP\times AP)\div\sqrt{(BP)^2+(AP)^2}$$ (see footnote 6)

$$AP=CP\times cot\ VH$$

The average number of teeth in contact in the axial plane is

$$N_a=FW/AP$$

The average length of contact lines may be obtained if the normal base pitch, line of action and face width are known. This is shown in *Fig.* 10. The equation for the average contact length is

$$CL_{ave}=(FW\times LA)/NBP$$

Since the maximum stress is obtained during that part of the cycle of rotation when the length of contact lines is a minimum, equations must be derived for the determination of this minimum contact length. *Fig.* 19 shows the zone of action for a set of gears when N_r is greater than one, but less than two, and N_a greater than one. It is apparent from the figure that minimum length of

(Text continued on Page 59)

[6] *Gear Geometry*, by Allan H. Candee

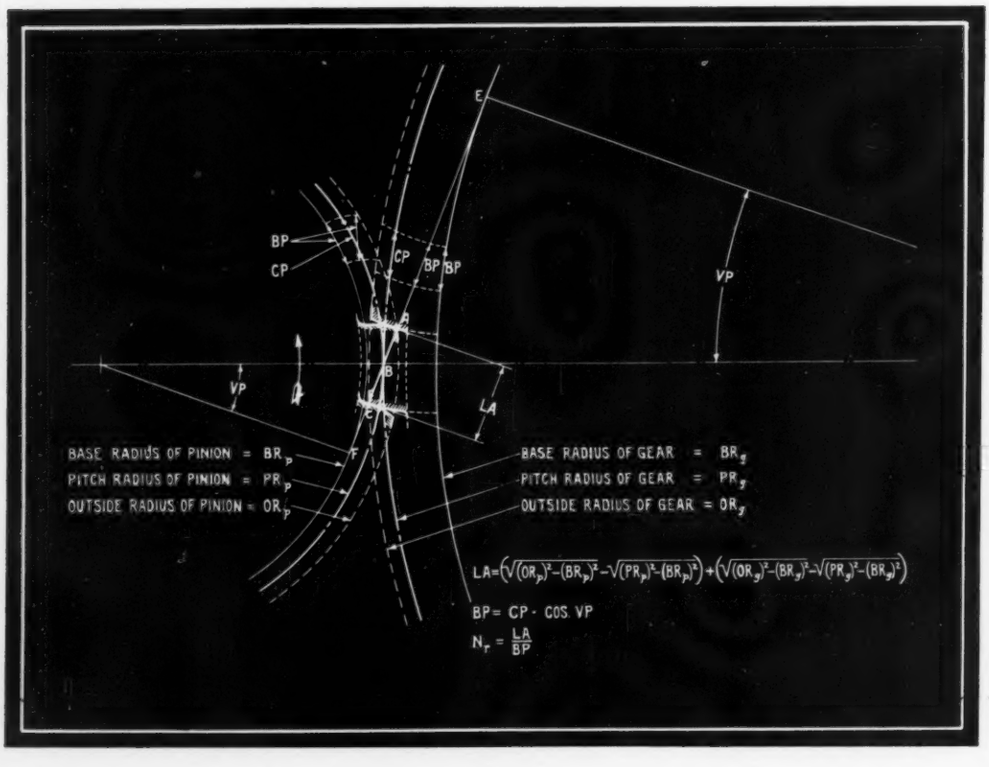


Fig. 11 — The line of action may be computed from simple trigonometric and geometric functions

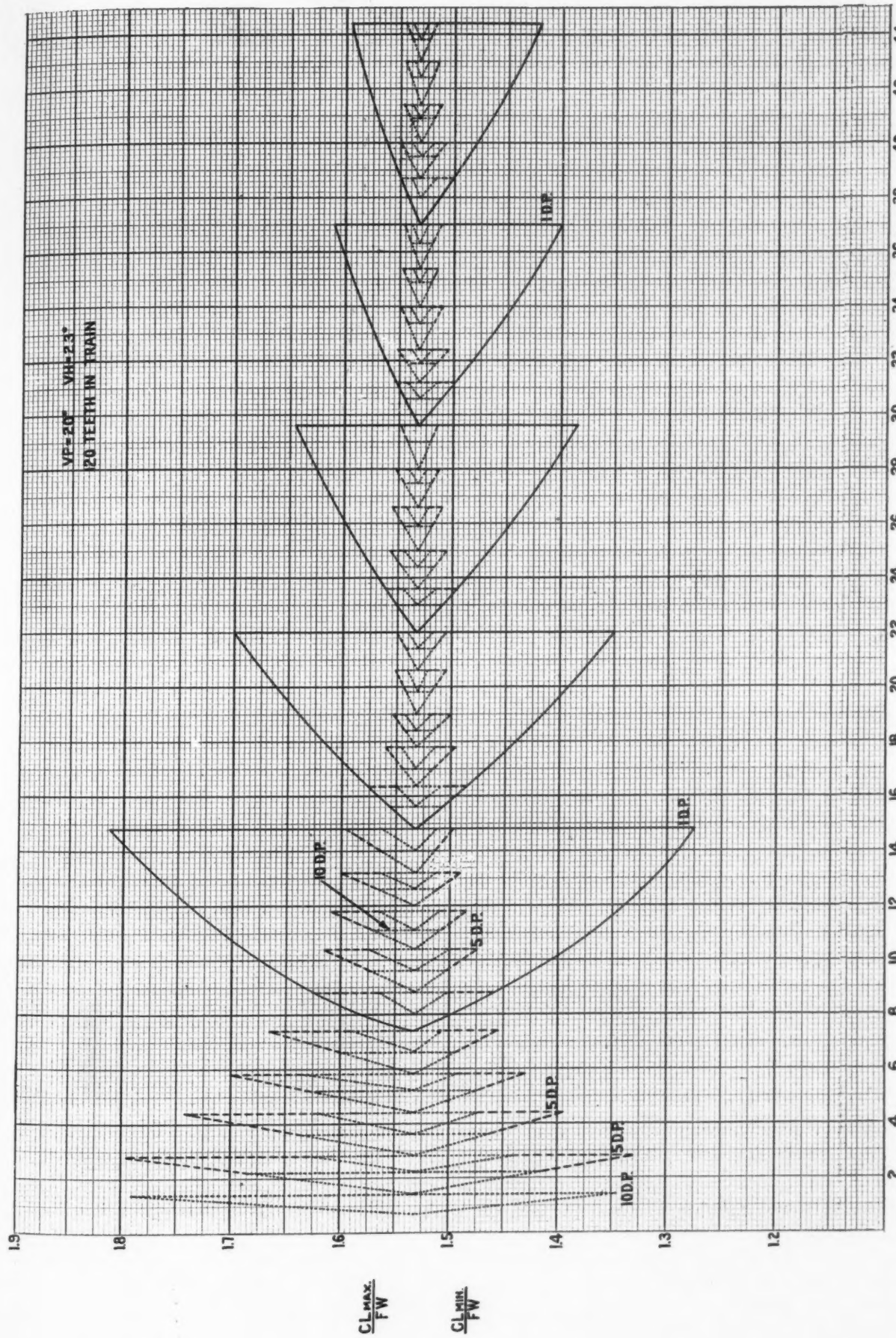


Fig. 12—Ratio of total contact length to face width varies with ratio of pitch to face width for a given tooth height, pressure angle, helix angle and number of teeth

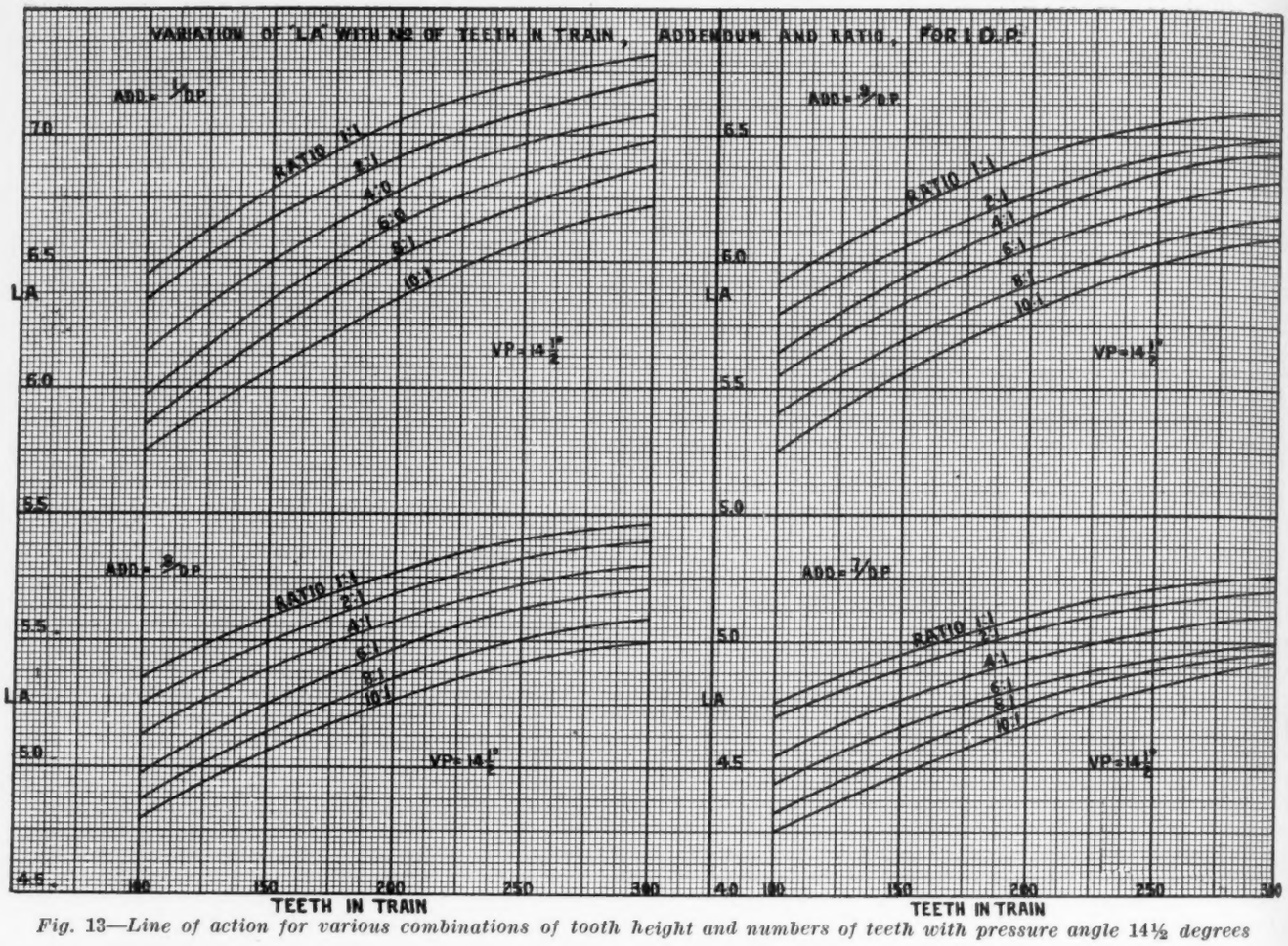


Fig. 13—Line of action for various combinations of tooth height and numbers of teeth with pressure angle 14½ degrees

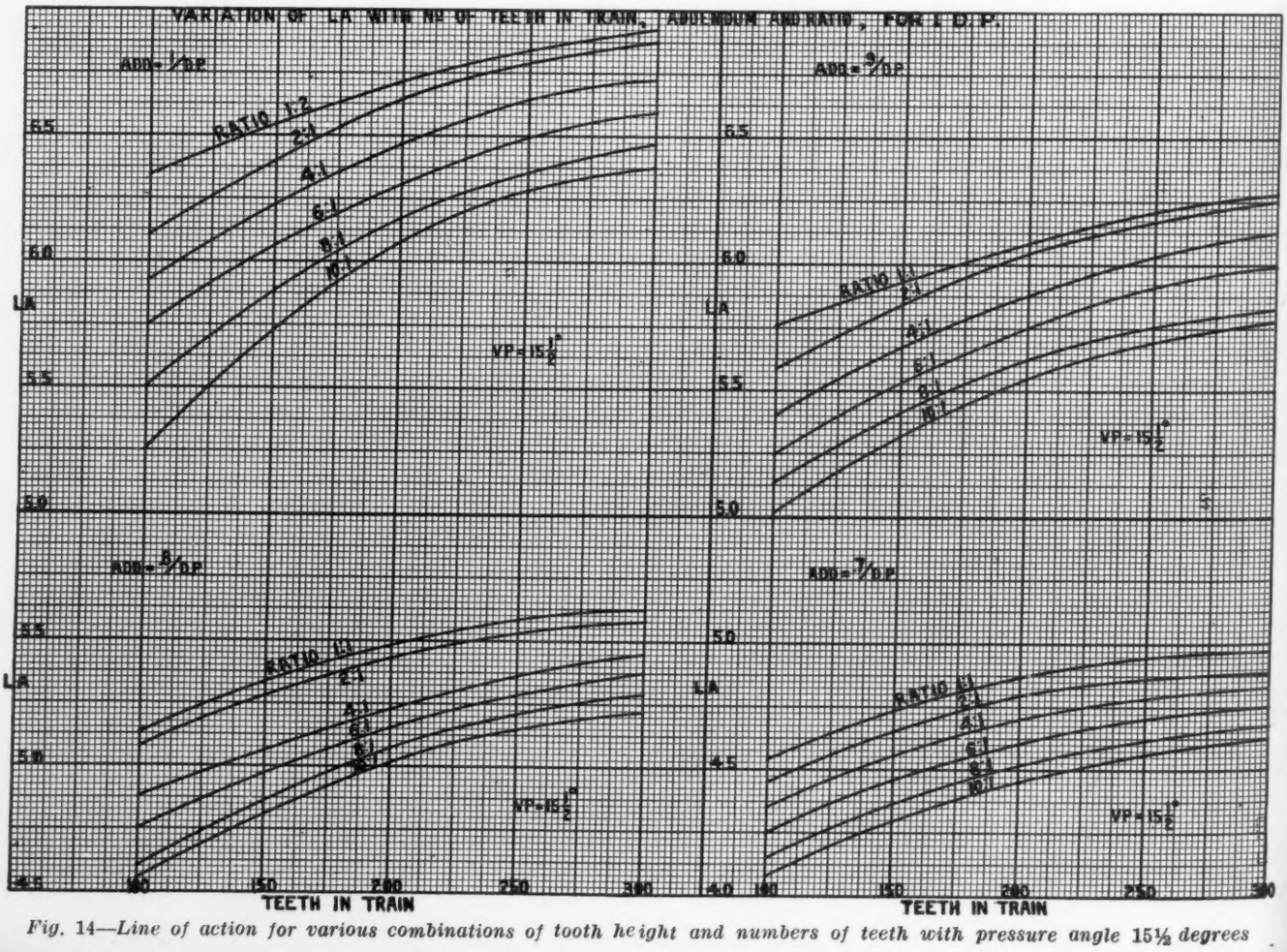


Fig. 14—Line of action for various combinations of tooth height and numbers of teeth with pressure angle 15½ degrees

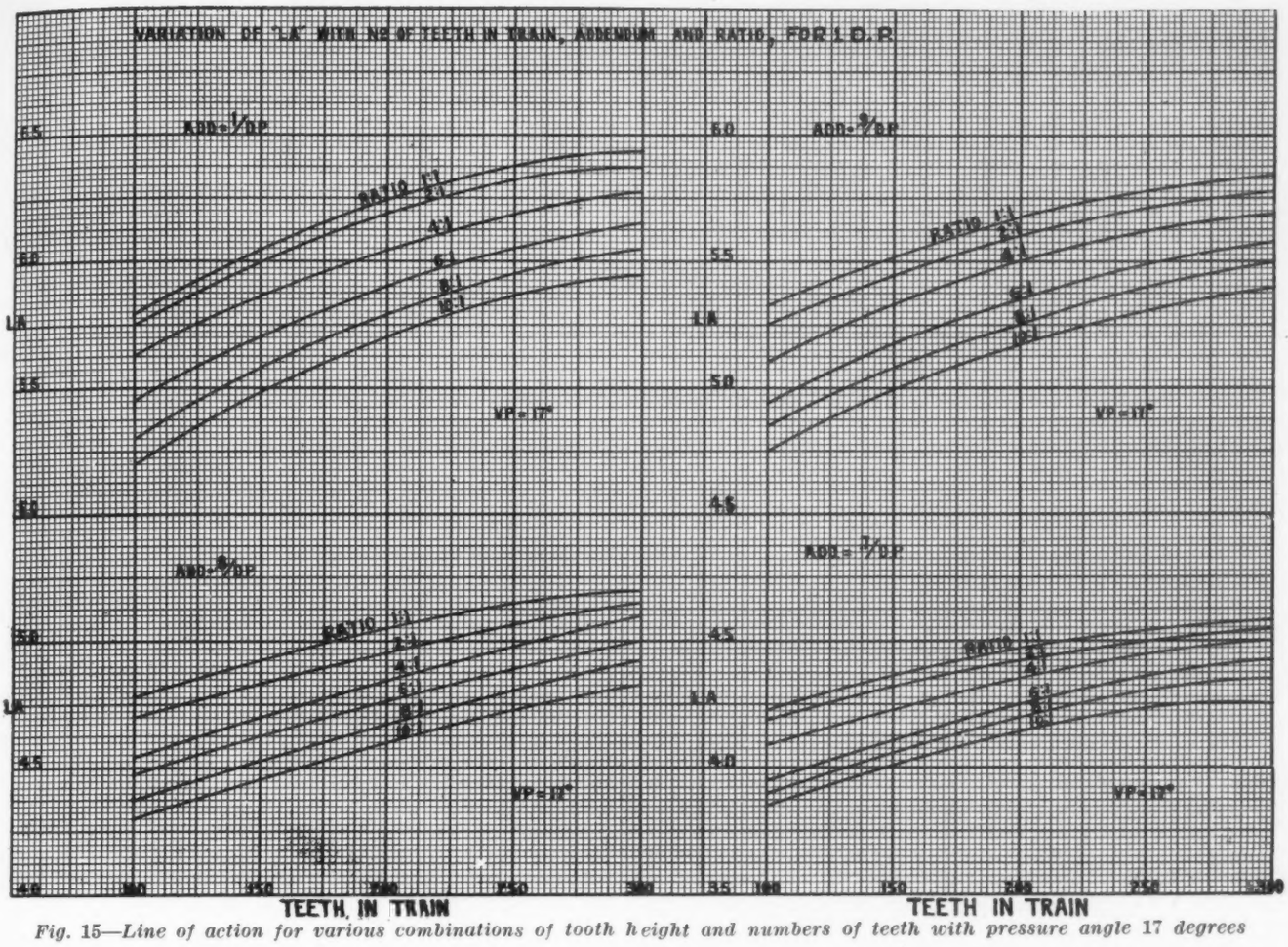


Fig. 15—Line of action for various combinations of tooth height and numbers of teeth with pressure angle 17 degrees

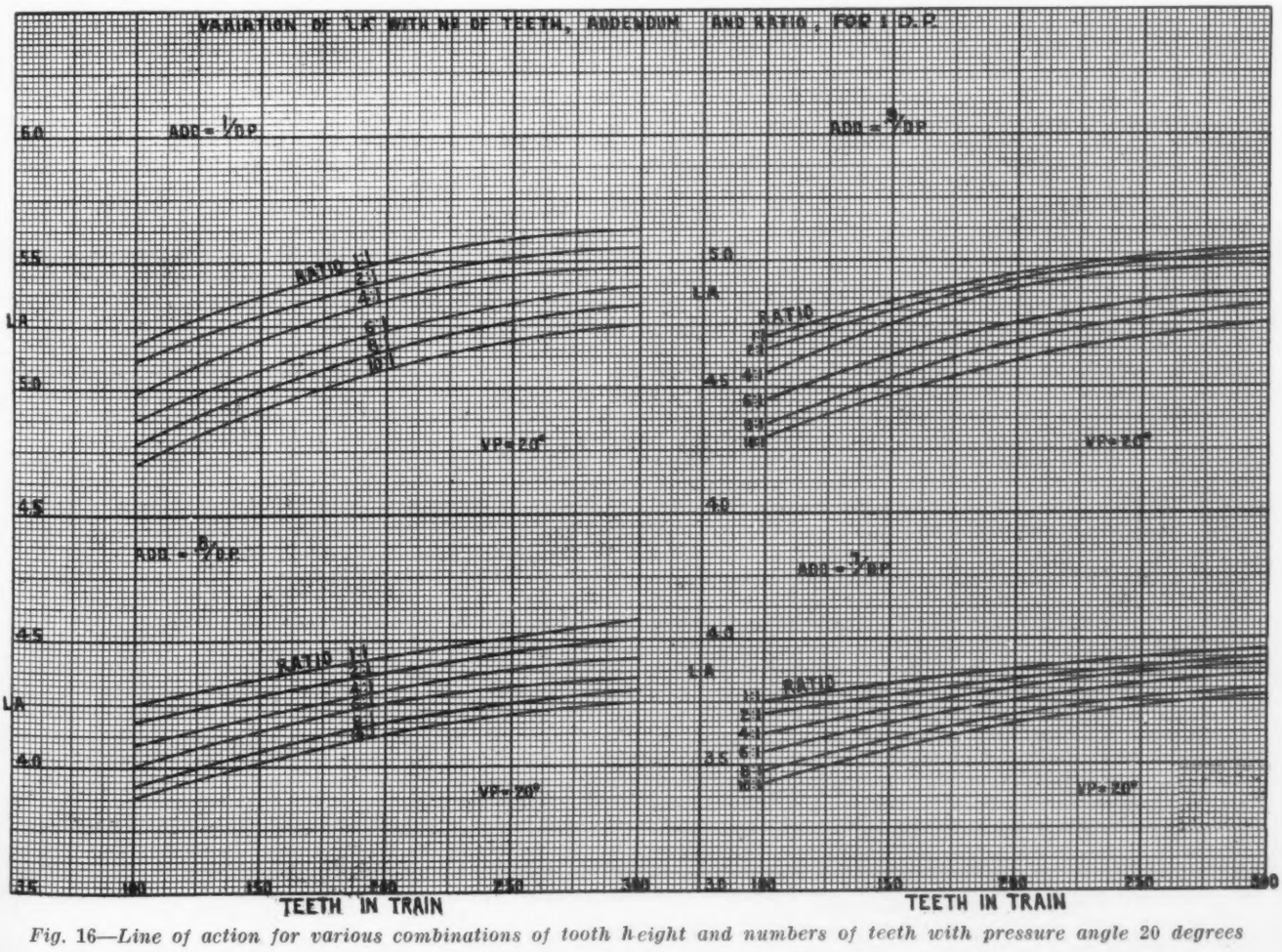


Fig. 16—Line of action for various combinations of tooth height and numbers of teeth with pressure angle 20 degrees

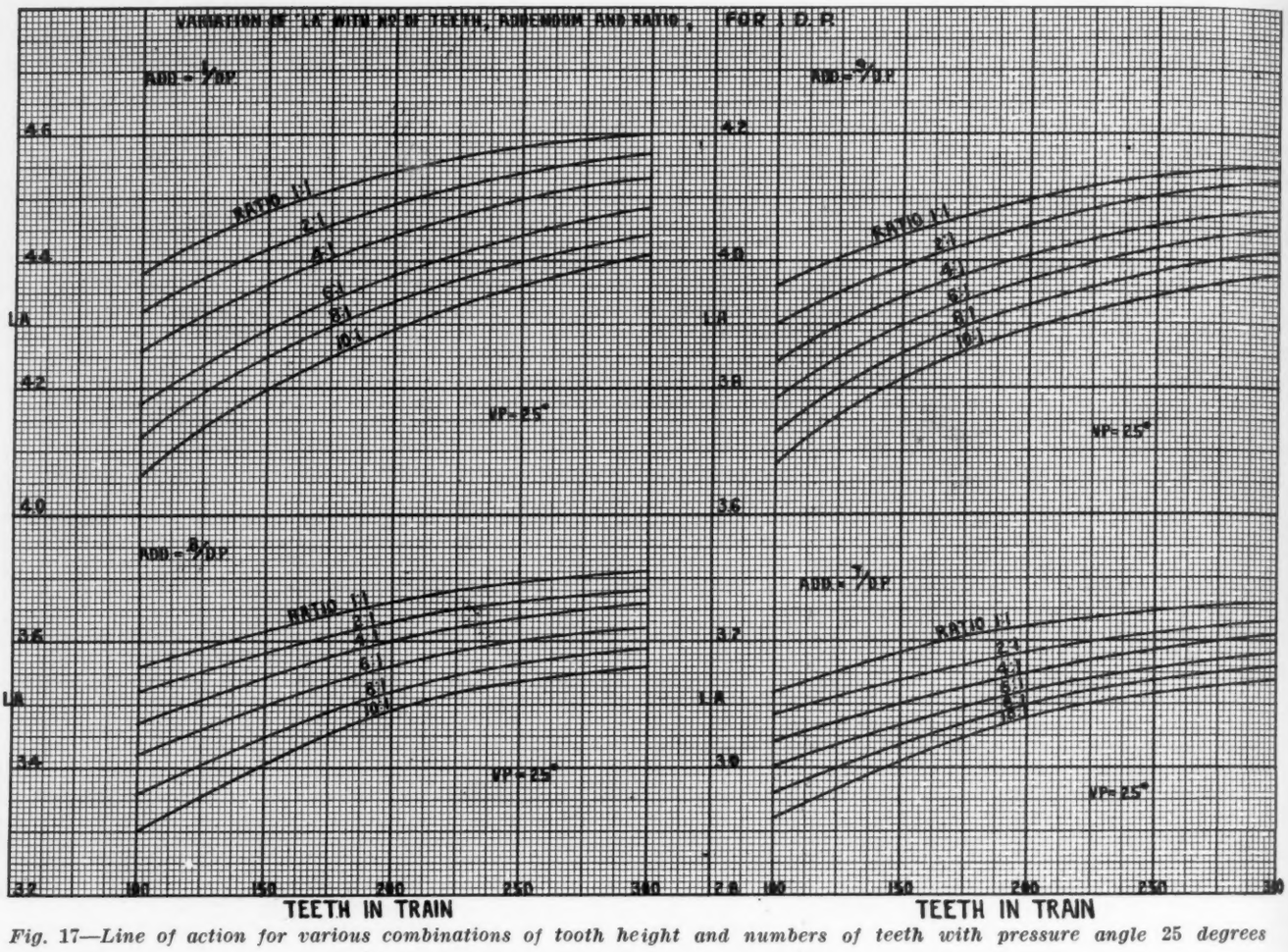


Fig. 17—Line of action for various combinations of tooth height and numbers of teeth with pressure angle 25 degrees

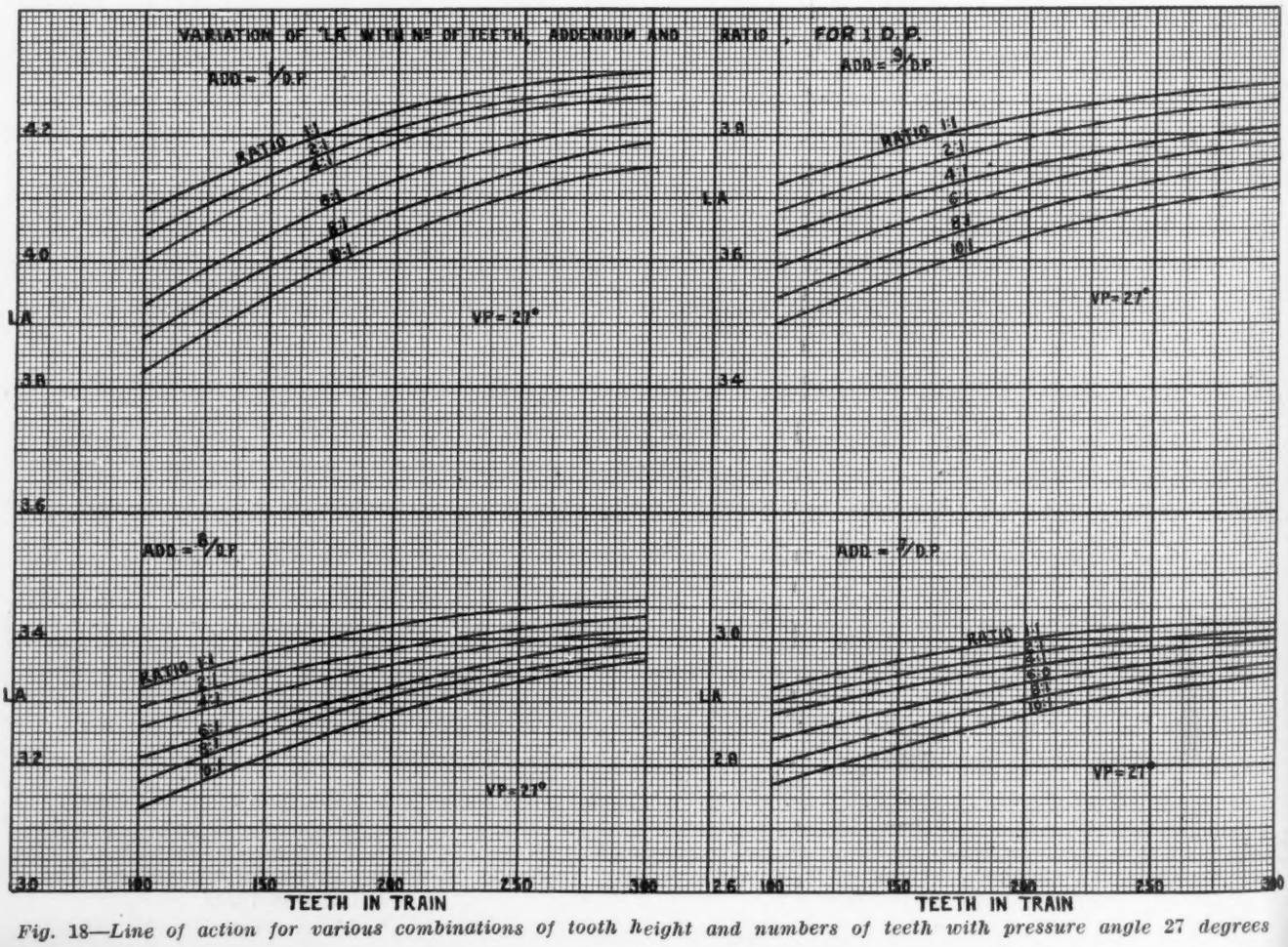


Fig. 18—Line of action for various combinations of tooth height and numbers of teeth with pressure angle 27 degrees

(*Text Continued from Page* 54)

contact lines is a function of both base and axial pitch, and the ratio of one to the other. Minimum contact occurs when a new tooth is about to engage. The equations given are for various relations of face width, axial pitch, base pitch and line of action.

Fig. 21 shows the zone of action and equations for a set of gears when N_r is greater than two, but less than three, and N_a greater than two. *Fig.* 20 shows the zone of action and equations for a set of gears when N_r is greater than two, but less than three, and N_a greater than one, but less than two. When either N_r or N_a is an integral number there is no variation in total length of the contact lines, and $CL_{min} = CL_{ave}$.

Fig. 22 shows the allowable pressure per inch of face obtained on a series of gears of various ratios and sizes based on computation of the exact minimum length of contact. The fact that a straight line can be drawn through these curves and that none of the points will vary more than 5 per cent either way indicates that exact determinations are not required if proper modifying factors can be found.

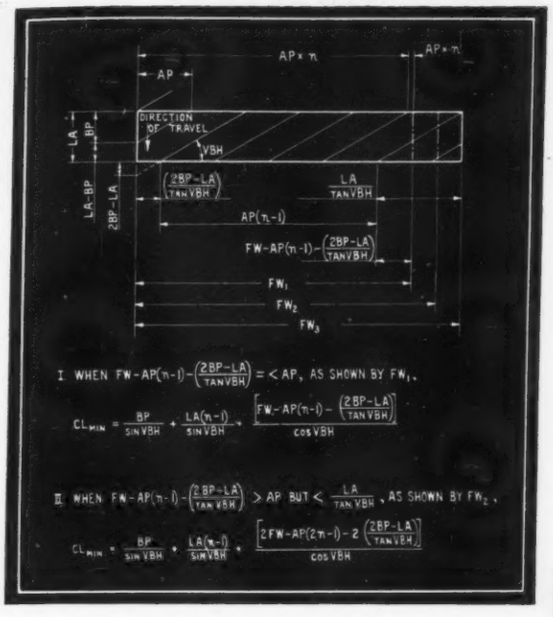
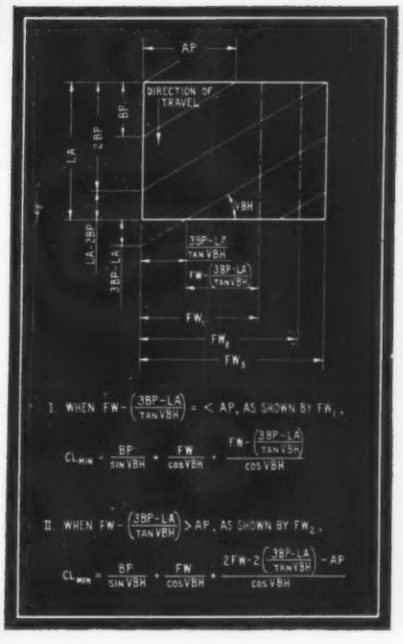


Fig. 19—(*Above*)—*Total minimum contact length equations for helical gears when number of teeth in contact in plane of rotation is greater than one but less than two, and number of teeth in contact in axial plane is greater than one*

Fig. 20—(*Right*)—*Total minimum contact length equations for helical gears when number of teeth in contact in plane of rotation is greater than two but less than three, and number of teeth in contact in axial plane is greater than one but less than two*

Fig. 21—(*Below*)—*Total minimum contact length equations for helical gears when number of teeth in contact in plane of rotation is greater than two but less than three, and number of teeth in contact in axial plane is greater than two*

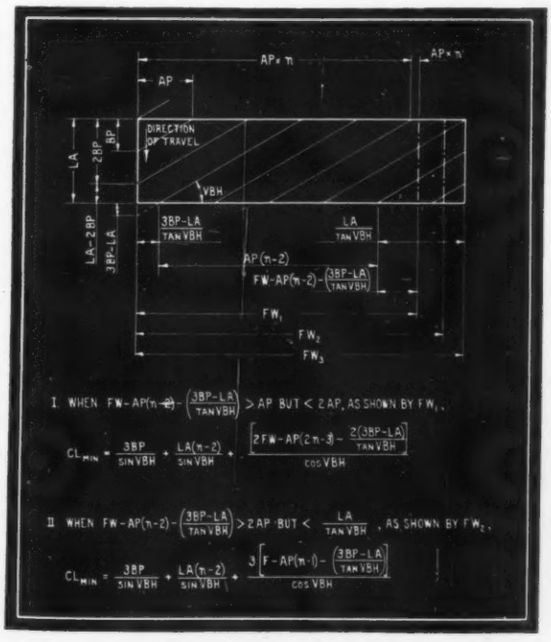


The length of the line of action of the rack would lead to a simple method of determining exact contact length if it could be used because *LA* could be determined by dividing the working depth by the sine of the pressure angle. A careful analysis shows that this would lead to serious error because the shortening effect of the outside circles varies considerably as shown in *Fig.* 23. The contact length of the rack is approximately 40 per cent greater than the actual for full depth teeth, 14½ degree pressure angle with 100 teeth in the train and only 12 per cent greater for 25 degree pressure angle teeth with a working depth of 1.4/D.P.

The curves of *Figs.* 13 - 18 give the length of the line of action for various combinations of pressure angle, tooth height and numbers of teeth. For high grade speed reducers CL_{min} has been found to average 0.95 CL_{ave}. For properly designed commercial helical gearing CL_{min} can be assumed as 0.9 CL_{ave}, thus obviating the necessity of tedious computation of the actual contact length. Since

$$CL_{ave} = (FW \times LA)/NBP;\ CL_{min} = 0.9(FW \times LA)/NBP$$

The remainder of this comprehensive analysis of horsepower ratings of helical and herringbone gearing will be presented in the July issue of MACHINE DESIGN.



Nomenclature

VBH angle, base, helix
VH angle, helix
VNP angle, normal pressure
VP angle, pressure
AP axial pitch
BR_p base radius of pinion
BR_g base radius of gear

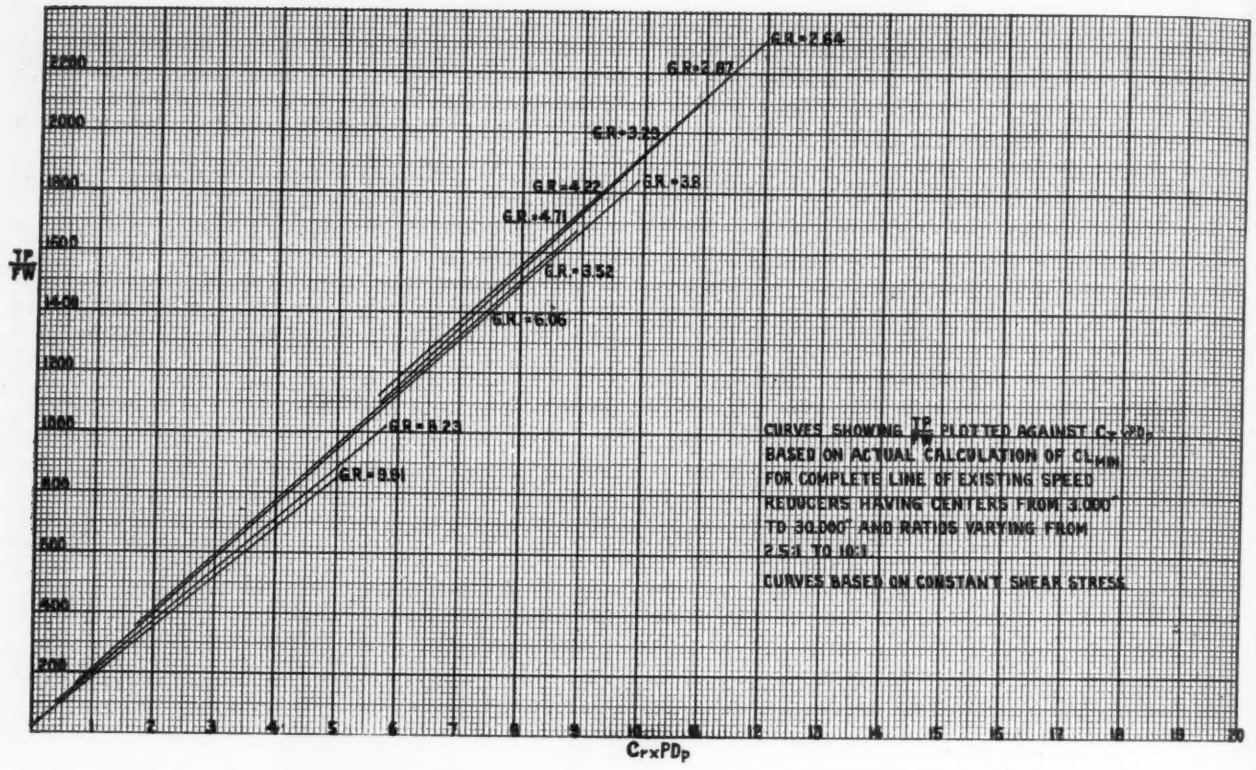


Fig. 22—Computation of loading on the basis of minimum contact length and constant shear stress indicates that average contact length may be used for well designed gears

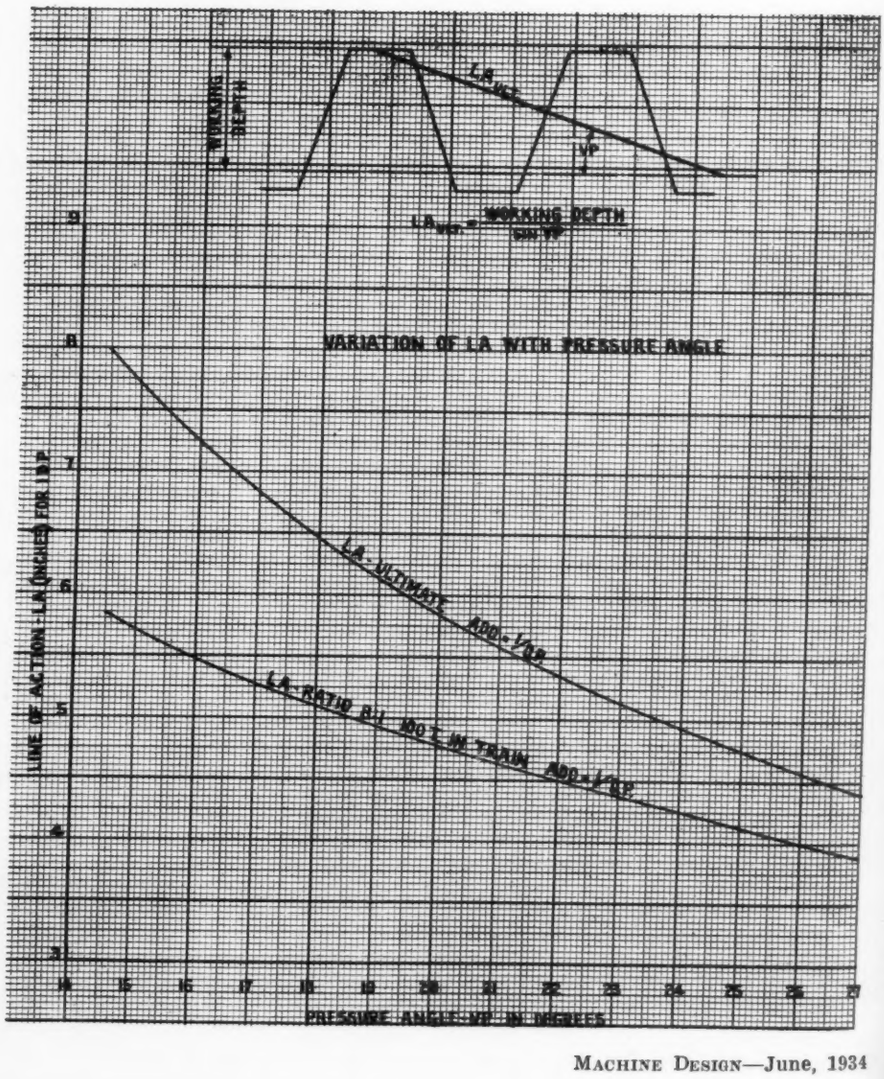

WORKING DEPTH
LA max
VP
LA max = WORKING DEPTH / SIN VP
VARIATION OF LA WITH PRESSURE ANGLE
LINE OF ACTION LA (INCHES) FOR 1 D.P.
LA ULTIMATE ADD 1/DP
LA RATIO 3:1 ADD 1 IN TENTH ADD 1/DP
PRESSURE ANGLE VP IN DEGREES

Fig. 23—(Right)—The relationship betwen the actual line of action and the ultimate length based on the working depth of rack is not constant

BP base pitch
CD_i center distance
CP circular pitch
CL_{ave} Contact length, total average
CL_{min} contact length, total minimum
CL_{max} contact length, total maximum
$\wedge$ contact and elasticity factor
DP diametral pitch
FW face width
Factors
C_q—contact length factor
C_c—curvature factor
C_i—combined inbuilt factor
C_m—material factor
C_r—ratio factor
C_s—service factor
C_v—velocity factor
f—factor of safety
GR gear ratio
HP horsepower
LA line of action
E_1, E_2 moduli of elasticity of cylinder 1 and 2
E_p, E_g moduli of elasticity of pinion and gear
NBP normal base pitch
N_p number of teeth in pinion
N_g number of teeth in gear
N_a average number of teeth in contact in axial plane
N_r average number of teeth in contact in plane of rotation
n number of full contacts in axial plane
Example: $N_a = 3.6, n = 3$
n' number of fractional contacts in axial plane
Example: $N_a = 3.6, n' = 0.6$
OR_p outside radius of pinion
OR_g outside radius of gears
PD_p pitch diameter of pinion
PD_g pitch diameter of gear
PR_p pitch radius of pinion
PR_g pitch radius of gear
v Poisson's ratio
P' pressure per unit length of roller
TP pressure, total tangential on teeth
TP_n pressure, total normal on contact lines
r_1, r_2 radii of cylinders 1 and 2
RPM revolutions per minute
S_e endurance limit stress for gears
S_{max} maximum specific compressive stress
S_s maximum shear stress below surface
$S_{s\,s}$ maximum shear stress at surface
Stress Components
${}_oX_x$ max. end stress at surface
${}_oY_y$ max. tangential stress at surface
${}_oZ_z$ max. compressive stress at surface
${}_sX_x$ max. end stress below surface
${}_sY_y$ max. tangential stress below surface
${}_sZ_z$ max. compressive stress below surface
V velocity at pitch circle—ft./min.
b width of flattened area

Gear Experts Discuss Schmitter's Analysis

Should Be a Future Standard

IN ACCORDANCE with your request, I have reviewed with much interest the advance copy of Mr. Schmitter's article on the design and capacity of helical and herringbone gears and I consider it a valuable contribution to the gear engineering field.

His arguments are well founded and his conclusion logical. In his reference to pitting, spalling, etc., he has diagnosed and defined the ills of gear teeth in such a way that the same can and should be accepted as a future standard.

—A. A. Ross,
General Electric Co.

Strength Calculations Unnecessary

MR. SCHMITTER'S article is a very valuable work; that he has devoted an enormous amount of time to it, is obvious; but the intelligent way he has handled a difficult subject is even more commendable. Besides having collected and interpreted much previous work done by others, he has done a large amount of new work which is useful in a practical sense and of great utility for purposes of reference, and it will influence and assist other investigators.

Many engineers of a practical turn of mind will object to the article as being in the main too academical and no doubt Mr. Schmitter is quite aware of the likelihood of such criticism. On the other hand he has attempted—after having dealt with the subject more or less academically—to show its practical applicability. To many it would appear very useful if he had left the academic considerations out of it; on the other hand the value of the article would be greatly reduced if he had not shown how he has arrived at his conclusions.

Such a paper deserves the greatest commendation, even though many parts of it are controversial.

Mr. Schmitter has dealt almost exclusively with the load carrying capacity of gears on the basis of durability rather than strength. Students of the subject have discovered that calculations for strength, providing certain simple rules are observed, are unnecessary and the time spent making them is wasted. It has been found that if a pinion does not contain over a certain number of teeth its resistance to breakage is far more than its resistance to wear; consequently, if a pinion has not less than a certain number of teeth, which is generally agreed to be somewhere between 30 and 40, no strength calculations need be made because if it is large enough and of a suitable material to be durable, it will be more than amply strong.

Fundamentally, the conclusions Mr. Schmitter has arrived at are that the load transmitting capacity of a pinion is in direct proportion to its diameter and its face width. This has been known for a long time but many engineers have hesitated or refused to believe it. Mr. Schmitter has shown in an able manner that it appears to be based on correct analysis.

The correctness of the results obtained from the factors which Mr. Schmitter has given depends, of course, on whether the factors are correct. Other students of the subject will undoubtedly question many of them and it would be dangerous as well as unwise to accept the values suggested as being correct. However, Mr. Schmitter has ventured to give many values which others have been afraid to do and he has at least given some figures which can be tried out and studied.

The application of the Hertz equation, which relates to cylinders in contact, to gear teeth has during the last few years become quite common and it seems to be useful and

of practical benefit. However, there is a decided difference between the action of teeth meshing together and the action of rollers bearing on each other. Up to the present, however, there seems no better way of dealing with the load carrying capacity of gear teeth excepting by the use of this equation. Probably in the course of time some better equation will be evolved.

The value for the factor C_m is debatable and many pages of arguments can be written about it, and in all probability there will be much discussion on this one factor. The same may be said regarding all the other factors with the exception, perhaps, of the ratio factor which is generally accepted as being correct.

The article attempts to provide equations which will enable the non-gear technician to determine the sizes of gears to transmit a given load or, alternatively, to determine the load carrying capacity of a given gear. It seems, however, doubtful whether this object has been achieved. It is always more difficult to specify suitable gears when the power and speed are specified than to determine what power gears will transmit when the gear proportions and the speed are specified, and it is more often the problem is in the first mentioned form.

I have found the following equation to be useful for general purposes:

$$C^2 W = \frac{HP(R+1)^2 315}{rpm.\, C_r.\, q}$$

where

C = center distance in inches
W = width of face in inches
R = ratio
C_r = ratio factor
q = the product of the various factors

such as Mr. Schmitter's C_m, C_v, C_i, C_c, C_q and many others which may be devised. The product of C^2W results in a number which is a measure or an indication of the load carrying capacity of a gear housing. When this number is obtained various alternatives of C and W can be easily read from the slide rule, and the most convenient or most suitable values can be chosen for each symbol. Moreover, it can be immediately ascertained which of a standard series is suitable for the load specified. This equation shows that other things being equal, the load carrying capacity of herringbone gears is in direct proportion to the square of the center distance multiplied by the width of face.

Service Factors Given

Regarding the factor q, it is practicable and usual to assume a value which has been determined by observation and experiment for the particular kind of service. These values are as follows:

Industrial steam turbine units for continuous service 0.6
Steam turbine units for warships 0.9
Heavy duty reciprocating pumps, continuous service, 0.7
Intermittent duty reciprocating pumps 0.9
Oil well pumping equipment 1.00
Cranes and hoisting machinery 1.5 to 2.85
Airplane engine gears for driving the propeller, transport work, 4.00 racing 6.5

It will be seen that the above factors not only combine all the various C factors dealt with Mr. Schmitter, but also the service factors which he has suggested. It is probable that in the present state of knowledge the combination of all these factors in this way is quite as accurate as taking them all separately; at least it is of more service in practice, but it is open to the objection that it leaves much to the designer's experience and judgment. It is necessary for him to choose material to suit the particular service. On the other hand, the factors are based on the most suitable material having regard to practical and economical considerations.

At one time it was expected any mechanical engineer could easily determine and specify gear sizes. Mr. Schmitter's article shows that considerable knowledge is desirable to obtain gears which are reliable and reasonably economical. It used to be quite a common experience that gears failed or were noisy, and in many other ways troublesome. The same can be said of other mechanical devices. Improvement in all mechanical devices has been made in the past, and will be made in the future, by utilizing all such data and information as Mr. Schmitter has made available. Further progress will be made by checking and possibly improving and amplifying the data which Mr. Schmitter has so ably developed and compiled.

—W. E. Sykes,
Farrell-Birmingham Co. Inc.

Method of Comparison Available

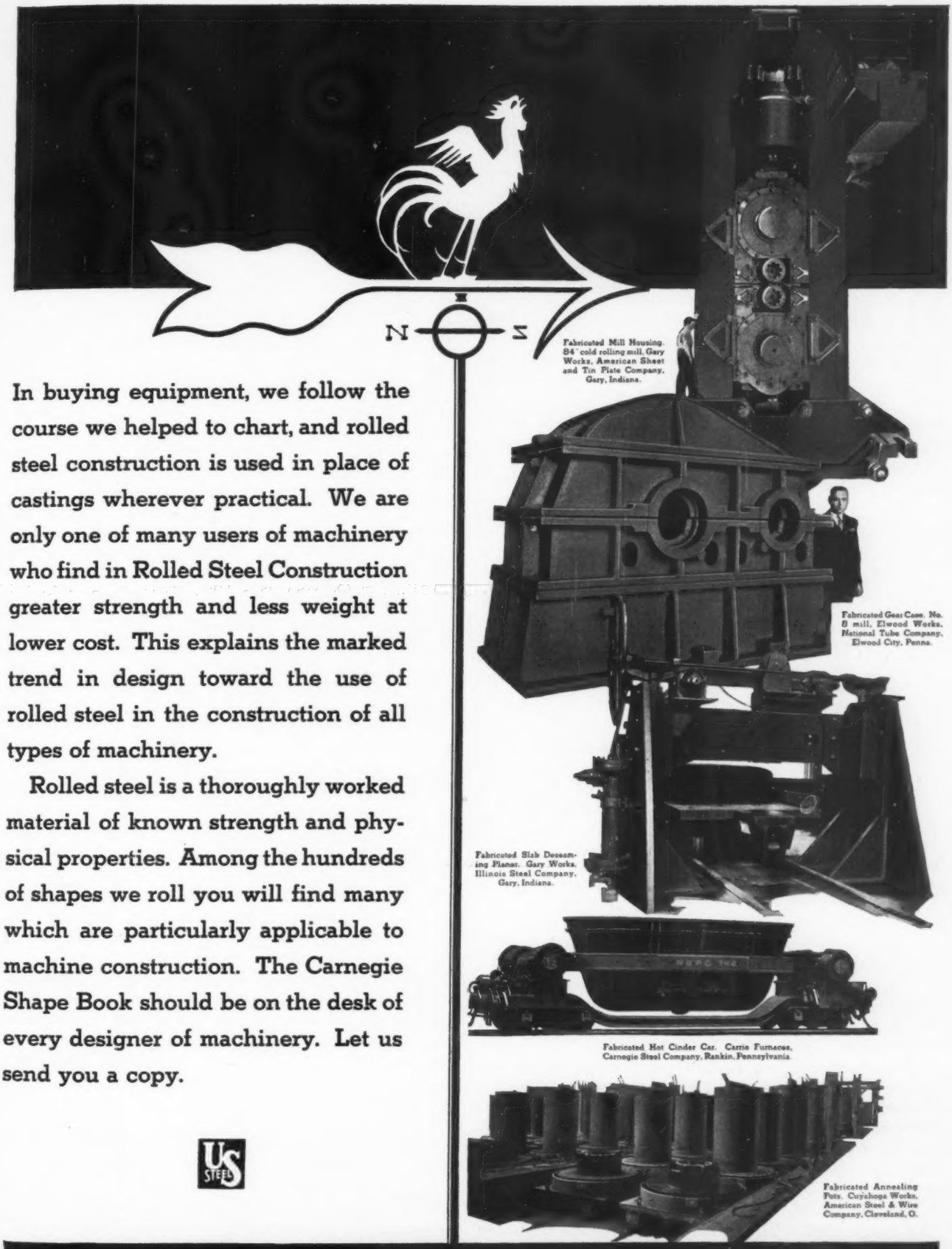
Mr. Schmitter has done a very good piece of work. With any such problem, it is always of great help to have a reasonably complete story written down, whether or not all of the details are perfect. With such a start, a consistent method of comparing results on widely different drives is available, and any corrections or modifications of the analysis may be made later as found necessary.

For one thing, the factor of accuracy plays a large part in the actual load-carrying capacity of the gears. The order of accuracy in one plant will vary from that in another depending upon many circumstances. If in one plant the great majority of the gears produced are relatively small, with occasional larger drives, the chances are that their order of accuracy on the smaller and more familiar product will be higher than on the larger and less familiar product. In such a case, each plant must determine for itself suitable limiting factors for use in determining the load-carrying ability of their product. The same will also be true if some unfamiliar material is specified.

As regards the influence of velocity, this is still an open question, and one person's opinion is probably as valuable as another's. I feel quite certain that the dynamic load is not directly proportional to the applied load, but this does not prevent me from using a velocity factor which is based on such an erroneous assumption where I have nothing better to offer at the time. The velocity factor given in the table, however, (to be published in the July issue) is one that has been established from experience with high speed gears (4000 feet per minute and greater) and I would anticipate that it would give too high a rating for lower speed gears, providing the stress values used were those suitable for the higher speed

Another factor which has received relatively little attention is that of the torsional deflection of wide faced pinions. If this torsional deflection is greater than the deformation of the tooth, full contact across the face cannot be secured. The load is probably never distributed evenly across the face of the gears. With the narrower faces such as used in standard reduction drives, this factor is of minor importance. With very large drives, however, where space is limited, and the face of the pinion may be several times its diameter, smaller unit loads must

(*Continued on Page* 89)



Fabricated Mill Housing. 84" cold rolling mill. Gary Works, American Sheet and Tin Plate Company, Gary, Indiana.

Fabricated Gear Case. No. 9 mill. Elwood Works, National Tube Company, Elwood City, Penna.

Fabricated Slab Deseaming Planer. Gary Works, Illinois Steel Company, Gary, Indiana.

Fabricated Hot Cinder Car. Carrie Furnaces, Carnegie Steel Company, Rankin, Pennsylvania.

Fabricated Annealing Pots. Cuyahoga Works, American Steel & Wire Company, Cleveland, O.

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USS

NOTEWORTHY PATENTS

MECHANISM to support citrus fruit and extract the juice by reaming is one of the principal features of the machine recently patented by Elof Nordstrom for the Hoffman Beverage Co., Newark, N. J. A conveyor belt 24, *Fig.* 1*A* carries the fruit which may be oranges, lemons, etc. from the supply mechanism (not shown). At the discharge end of each conveyor channel is a guide trough for directing the fruits to the pockets of a corresponding cutter holder

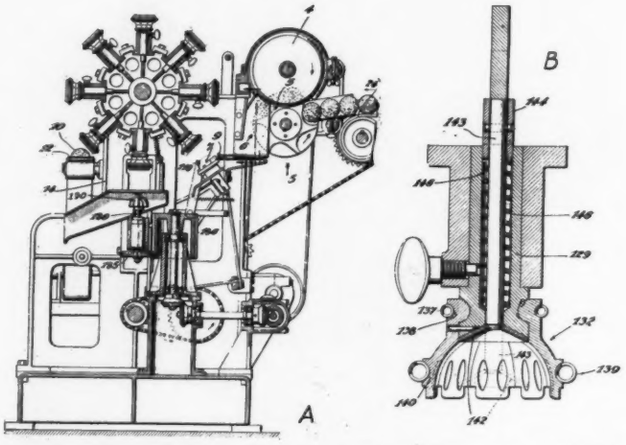

Fig. 1—*Citrus fruit, carried in chucks, is reamed to extract juice and then is ejected automatically*

or wheel 5. The pockets are arranged so that as they rise each one takes the foremost fruit from the corresponding guide trough and carries it up to contact with the knife 4, which cuts through the center of the fruit.

For each cutting support and knife a pair of guide plates 6 is arranged to receive the severed fruit halves, guiding and directing them for further movement. Pushers 9 are moved rearward in the feeding direction to move a row of fruit halves simultaneously from the supporting plates to the transfer holders 7 which at that time are positioned adjacent the rear edges of the guide plates to receive them. As the transfer frame swings up and rearward to place the fruit halves in the chucks carried on a turret the cross rail 110 moves to an idle position where it does not interfere.

Chuck fingers 132, *Fig.* 1*B*, normally are held in closed position with their sides in contact, by a helical spring 137 located in an upper channel 138. Another helical spring 139 is located in a similar channel 140 adjacent the lower ends of the fingers. Each chuck also includes an ejector for the reamed skin. This comprises a cupped disk 142 located in the upper portion of the chuck socket and carried by the outer end of stem 143 which reciprocates in sleeve 129. A helical spring 146 within the sleeve engages the lower end of the sleeve counterbore and collar 144 to retain the ejector in retracted position as shown at *B*, *Fig.* 1.

The head 190 for reaming the fruit to extract the juice is mounted on the uper end of shaft 180, *Fig* 1 *A*. Gear 185 engages a main driving gear 186 which has a wide face so that the driven gear remains in mesh during the vertical travel of the carriage as it rises to ream the fruit. After this is completed the turret is indexed and an ejector bar acts on stem 143 to deposit the reamed skins 10 on the carry-off conveyor 12.

The patent is designated 1,922,818.

REVERSING mechanism in which changes have been embodied to facilitate simple and inexpensive construction as well as to make assembly and disassembly easier, is covered by a new patent. The inventor is Walter C. Ware and Evans Stamping & Plating Co., Taunton, Mass., is assignee. Casing 24, *Fig.* 2, is entirely open at its rear end or the end remote from the integral front wall 26 so that it can be machined internally by a simple operation. A brake band 28, encircling the casing, is designed to engage and hold it stationary to set the mechanism for reverse drive of driven shaft 12. Intermeshing large and small pinion gears 54 and 56 supported on pinion pins, are in mesh with driving and driven spur gears 16 and 20 to effect reverse drive when the casing is held stationary.

The mechanism to secure forward drive includes a set of clutch plates 58 disposed immediately at the rear of end plate 30. A pressure plate 64 axially loose but having a driving connection with driven hub 18 is designed to move

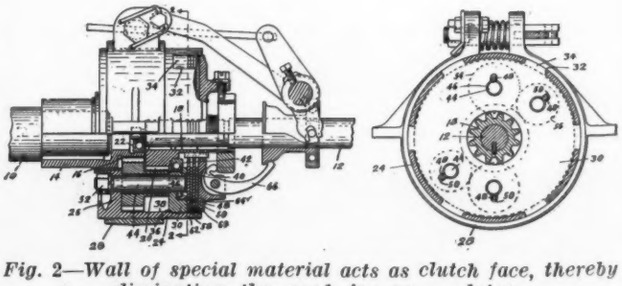

Fig. 2—*Wall of special material acts as clutch face, thereby eliminating the need for wear plates*

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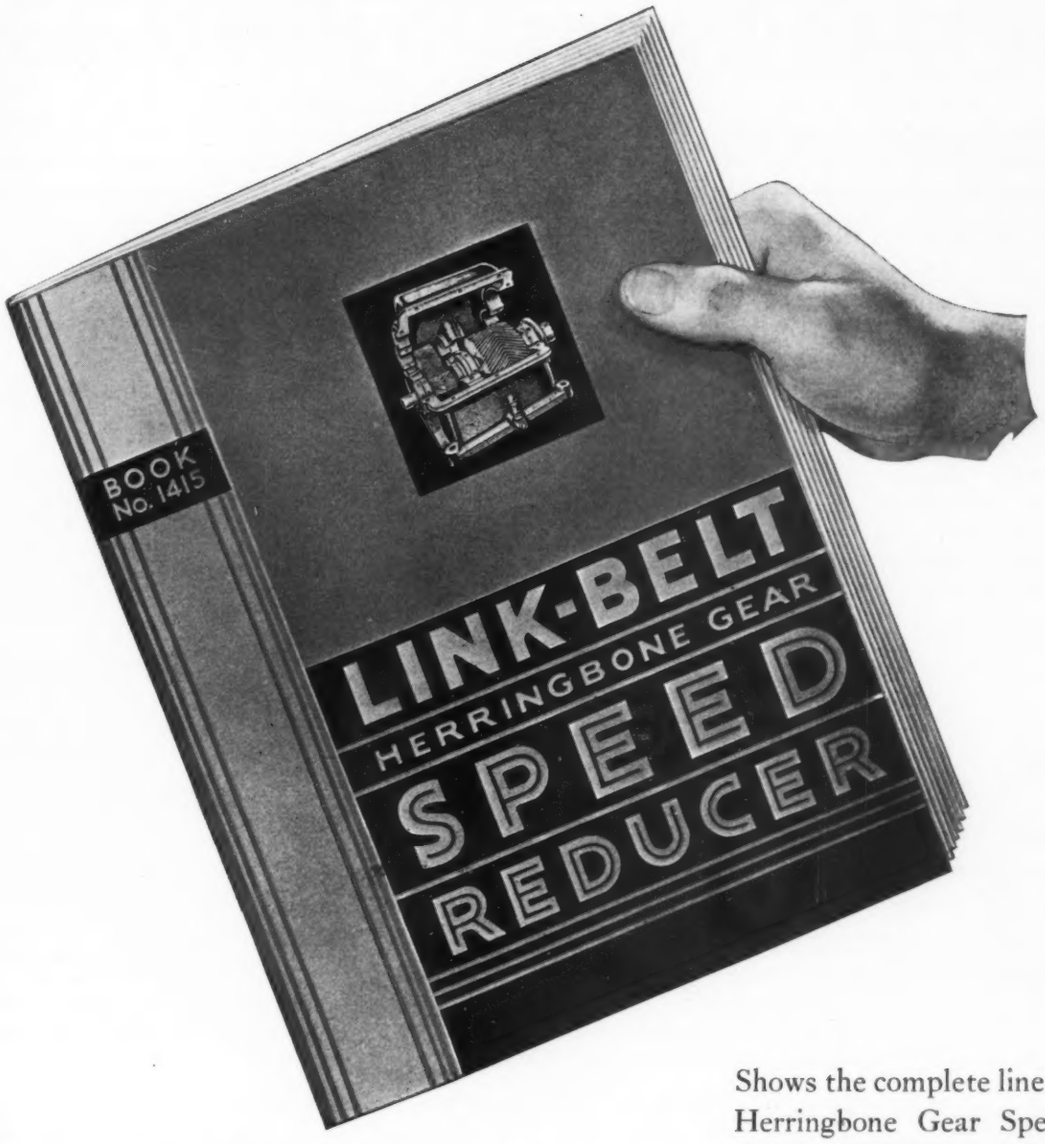

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Shows the complete line of Link-Belt Herringbone Gear Speed Reducers (Sykes tooth form). Gives complete data for selecting the correct unit for any requirement.

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the plates together and against end wall 30 by more or less common clutch operating mechanism 66 to set the reversing mechanism for forward drive.

In earlier construction of the reversing mechanism wherein the rear wall was integral with the casing and composed of the same material, it was found desirable to interpose a wear plate between the end wall and the foremost clutch plate of the driven shaft as the material of the end wall could not be provided economically with wearing qualities suitable for clutching service. With the present construction however, the separate end wall 30 is composed of a material that is different from that of the casing 24 and is adaptable for clutching service, thereby permitting the wear plates to be eliminated. With the end wall functioning as a clutch face, expense is saved and axial length of the gearing shortened appreciably.

Number of the patent is 1,955,662.

FOR automatically transferring articles from one conveyor to another, Edwin E. Milner has designed a leer feeder. The patent for this mechanism has been assigned to Ball Bros Co., Muncie, Ind. Belt 80 is driven continuously, receiving ware such as article 13, *Fig.* 3, from any suitable supply and carrying it into a leer in front of a pusher bar 87. Upon a predetermined movement of belt 80 a pin 118 actuates lever 117, which in turn actuates valve 116′ to admit air to casing 110. This drives valve 114 to a left-hand position to admit air to the right-hand end of cylinder 94.

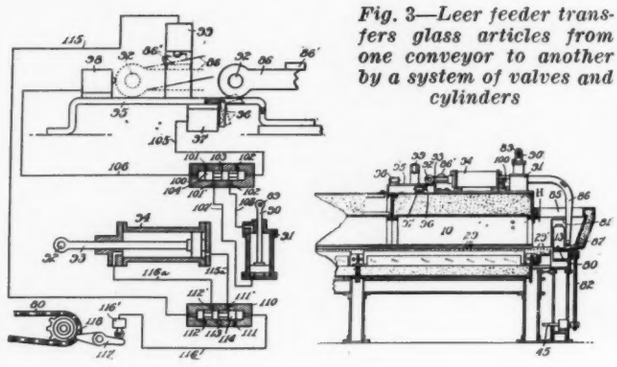


Fig. 3—Leer feeder transfers glass articles from one conveyor to another by a system of valves and cylinders

Rod 93 moves to the left, bar 87 sweeping the ware onto rollers 29′. Because these as well as rollers 29 are in rotation, the ware is advanced through the tunnel 10 of the leer. As finger 86 moves to the left it passes under valve 99 without actuating it and at the end of its movement in this direction, opens valve 98 to admit fluid under pressure to casing 100 to drive valve 104 to the right-hand position, thus admitting air to the lower end of cylinder 91.

This lifts the hangers pivoted at 89 on piston rod 90, raising pusher bar 87 above the level tops of ware 13 which is arriving on belt 80. When rod 90 reaches the upper end of its stroke a pad 86′ on finger 86 opens valve 99 and thus admits air to the left-hand end of casing 110 to drive valve 114 to the right, thereby venting the right-hand end of cylinder 94 and admitting air to its left-hand end. As this takes place pusher bar 87 is carried back over the tops of the arriving ware. When finger 86 arrives at the rear or right-hand end of its stroke it engages trigger 96 to open valve 97 to admit fluid under pressure to casing 100 to drive valve 104 to the left, admitting air to the upper end of cylinder 91 and lowering bar 87 to the starting point.

No. 1,952,680 identifies the patent.

Review of Noteworthy Patents

Other patents pertaining to design are described briefly as follows:

DOUGH SLICING APPARATUS—1,959,303. This machine embodies cutting knives and means for lubricating and cleaning the knives. Another feature comprises means for removing a sliced object from a supporting table where it is sliced, on to other means to facilitate removal of sliced objects from the machine. Assigned to Perkins Machine & Gear Co., West Springfield, Mass.

HUNT AND LAG PREVENTING MEANS—1,959,144. Covered by this patent is an invention relating to devices for transmitting the major rotary movements of a shaft subject to hunting or rapid oscillations of relatively small amplitude without introducing any lag in the major transmitted movements. Assigned to Sperry Gyroscope Co. Inc., Brooklyn, N. Y.

TORQUE MOTOR BRAKE CONTROL—1,928,293. One of the principal objects of the invention is to provide a braking mechanism in which the torque motor, when operated to release the brakes, is brought to a stop in a smooth and gradual manner and without heavy impacts or shocks on the torque motor or on any of the instrumentalities of the mechanism. Assigned to Harnischfeger Corp., Milwaukee.

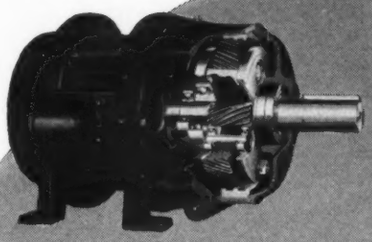
DEFROSTING MECHANISM—1,952,050. The invention covered by this patent, relates to the temperature control of automatic refrigerators and particularly to mechanism whereby the flow of operating fluid, such as electric current or gas to the motor or boiler of the refrigerator is stopped when certain conditions occur or are desired. Assigned to Charles J. Tagliabue Mfg. Co., Brooklyn, N. Y.

MACHINE FOR MAKING CAN OPENING KEYS—1,954,791. An important object of this invention is the provision of a series of wire feeding and holding devices for retaining control of the wire at all times during the key forming operation. Assigned to American Can Co., New York.

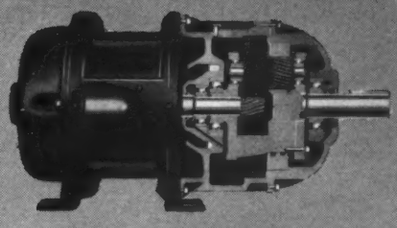
TENSION DEVICE—1,955,666. This invention relates to winding machines for winding cops, cones, thread, etc. The design provides a differential tension device adapted to operate automatically to maintain a uniform tension on the yarn during the winding. Assigned to Universal Winding Co., Boston.

SOUND in every detail of design and manufacture

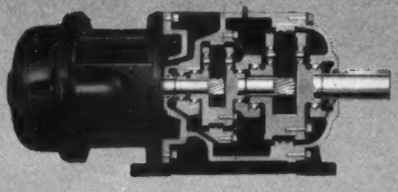
G-E gear-motors provide long life and reliable performance



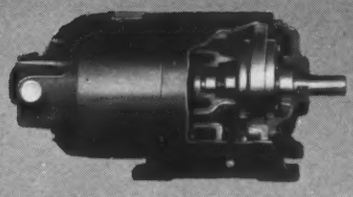
Cutaway view of G-E gear-motor, typical of 100-series gears in single-reduction units; output-shaft speeds, 600 to 154 rpm.



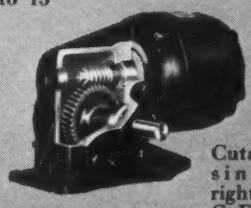
Cutaway view of G-E gear-motor, typical of 200-series gears in compound-reduction units; output-shaft speeds, 145 to 71 rpm.



Cutaway view of G-E gear-motor, typical of 300-series gears in double-reduction units; output-shaft speeds, 63.5 to 13 rpm.



Cutaway view of single-planetary, concentric-shaft, fractional-horsepower G-E gear-motor; 500 to 98 rpm.



Cutaway view of single-worm, right-angle-shaft, G-E fractional-horsepower gear-motor; 197 to 11 rpm.

STANDARD RATINGS

600 to 11 Rpm.

Polyphase, 1/6 to 75 hp.

Single-phase, 1/6 to 5 hp.

Direct-current, 1/6 to 7 1/2 hp.

Other speeds and horsepower ratings can be furnished on request. Mechanical modifications, such as flange mounting, vertical motors, etc., are available.

Speed adjustment can be obtained by selecting a motor with the requisite electrical characteristics.

YEARS of experience in the manufacture and application of gear reducers have established certain engineering principles which are the best assurance of a sound product and a long life of successful performance.

It is upon these principles that G-E gear-motors are designed, manufactured, and applied.

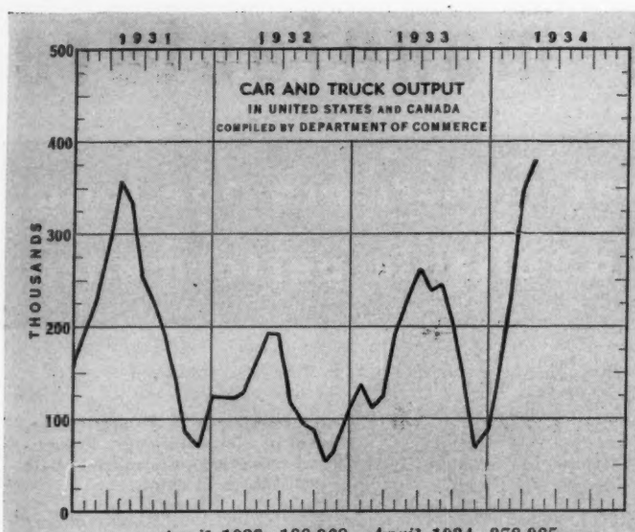
G-E gear-motors—a combination of motor with helical-planetary-type reduction gears—are available in a wide range of sizes and in a number of electrical and mechanical modifications to meet properly your requirements for low-speed drives.

G-E gear-motors provide full-horsepower output at the low-speed shaft. Their inherent characteristics—compactness, efficiency, ease of mounting, simplicity—make them the preferred electric drive wherever low speed is desired.

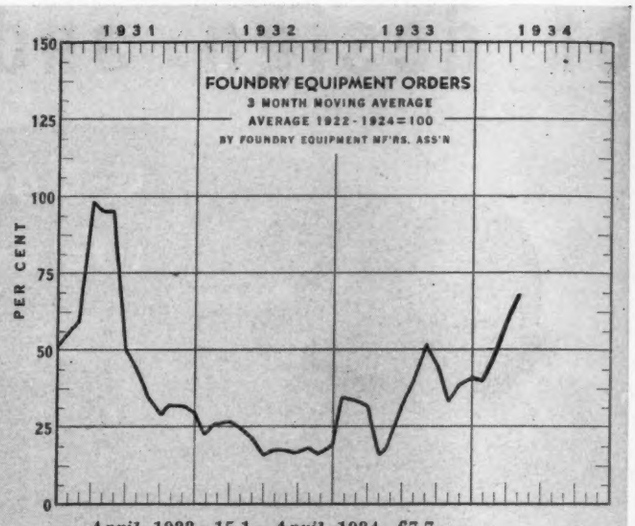
020-66

GENERAL ELECTRIC

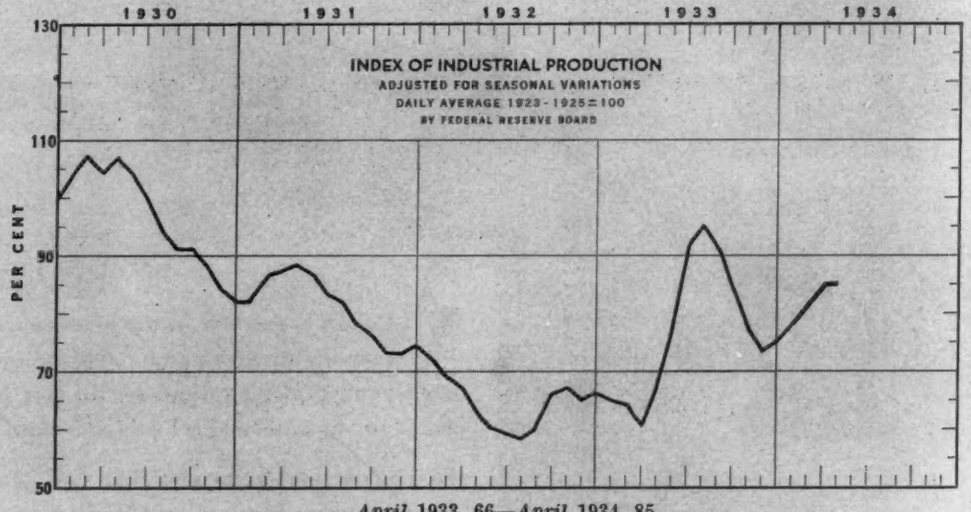
How Is Business?



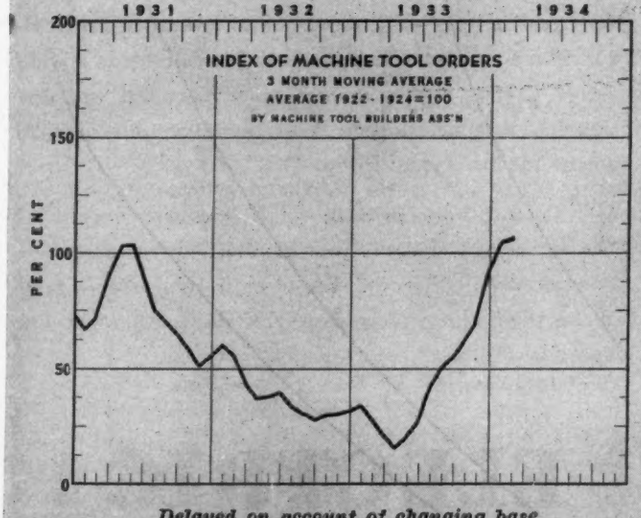
April 1933, 188,968—*April* 1934, 378,985



April 1933, 15.1—*April* 1934, 67.7



April 1933, 66—*April* 1934, 85



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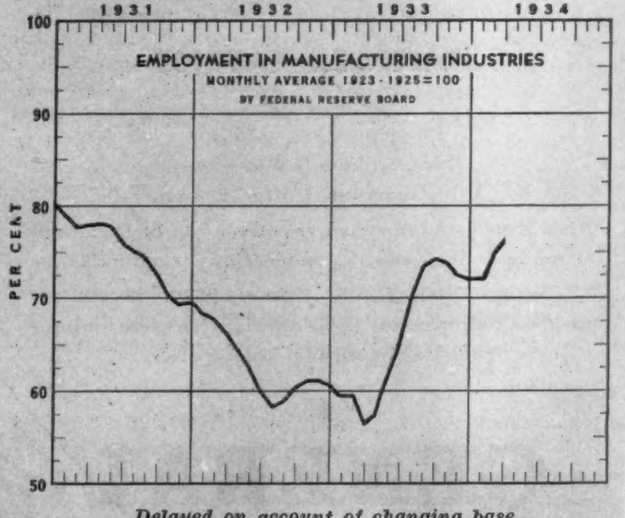

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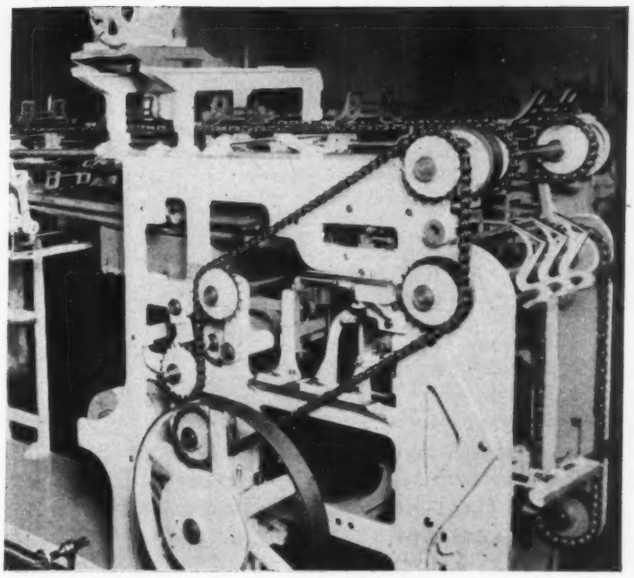


TRY THESE QUESTIONS ON *YOUR* DRIVES

Here are 9 questions which apply to every power transmission, conveying or elevating drive you install in your product or in your plant. If *your* answers are not entirely satisfactory, check those you would be able to give if you were using Baldwin-Duckworth Machine Finished Roller Chain Drives.

Modern machine design calls for many types of power transmission mediums. It calls for drives that will operate positively on individual machines — conveying, elevating, loading, wrapping — to mention a few. Today one must choose the best from among a number of good driving mediums, not always an easy job. Yet, the choice may be somewhat simplified by reading these questions, and the Baldwin-Duckworth answers.

BALDWIN-DUCKWORTH CHAIN CORPORATION
365 Plainfield Street, Springfield, Mass.

That TROUBLESOME DRIVE ?

1. **Is it slipless?**
2. **Is it positive in operation?**
3. **Is it affected by ordinary moisture or atmospheric conditions?**
4. **Does it require frequent adjustments?**
5. **Is it shock absorbing?**
6. **Will it operate at high efficiency?**
7. **Can it be subjected to overloads?**
8. **What about quick repairs?**
9. **What about first cost? Maintenance costs?**

BALDWIN-DUCKWORTH *ANSWERS*

1. **Machine Finished Roller Chain Drives cannot slip—this prevents power losses.**
2. **Yes—if it's Roller Chain.**
3. **Not if it's Machine Finished Roller Chain.**
4. **Adjustments are seldom necessary with Machine Finished Roller Chain Drives.**
5. **"Positively" is Baldwin-Duckworth's answer.**
6. **98.3% efficiency is possible through the use of Machine Finished Roller Chain Drives.**
7. **Machine Finished Roller Chains are repeatedly subject to momentary overloads of 100% without harm.**
8. **New repair links can be installed in a few moments.**
9. **Baldwin-Duckworth first cost is moderate, maintenance cost at a minimum, ultimate cost is less.**

BALDWIN DUCKWORTH

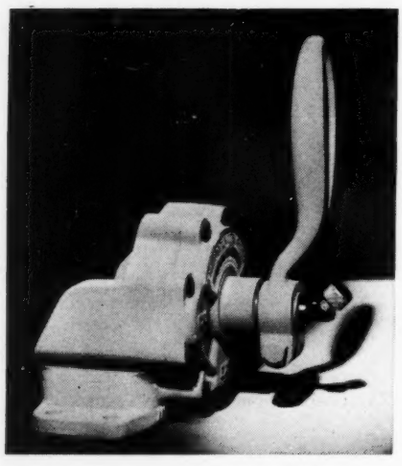
Factories **at Springfield and Worcester, Mass.**

NEW MATERIALS AND PARTS

Valve Has Balanced Disk Pressure

BALANCED disk pressure throughout contact area is attained by arrangement of seat ports and disk pockets in the new manifold valve introduced by Hanna Engineering Works, 1765 Elston avenue, Chicago. Between reversals the valve disk rotates on two eccentric points while in contact with its seat. The valve disk is seated on the valve bonnet and therefore when renewals are necessary they can be made without removing the valve body from the pipe line or disturbing pipe connections.

This valve, shown herewith, is particularly adaptable to those applications where a number of valves must be located close together. The necessary piping is possible without interference with the piping of adjoining valves.



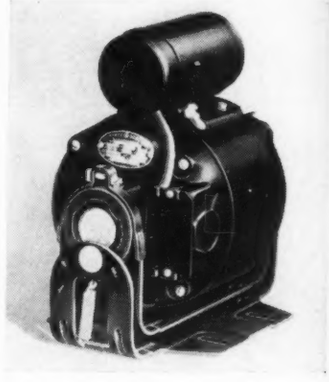
Between reversals the valve disk rotates on two eccentric points while in contact with its seat

---

Introduces Quiet Capacitor Motor

NOISE caused by torque vibration and by endwise and radial out-of-balance is eliminated in the design of the new capacitor motor introduced by General Electric Co., Schenectady, N. Y. This motor, shown herewith, is mounted in live rubber which is treated to make it impervious to oil. Other features include an end-play silencer eliminating noise by cushioning rotor oscillations in either direction; an optional belt tightener which maintains sufficient tension, proportional to the load, to prevent slippage; a cast rotor of solid aluminum formed by high pressure casting; large oil capacity with improved, wool-packed bearing; and oil throwers and returns which conserve the lubricant.

Motor is mounted in live rubber which is treated to make it impervious to the action of oil

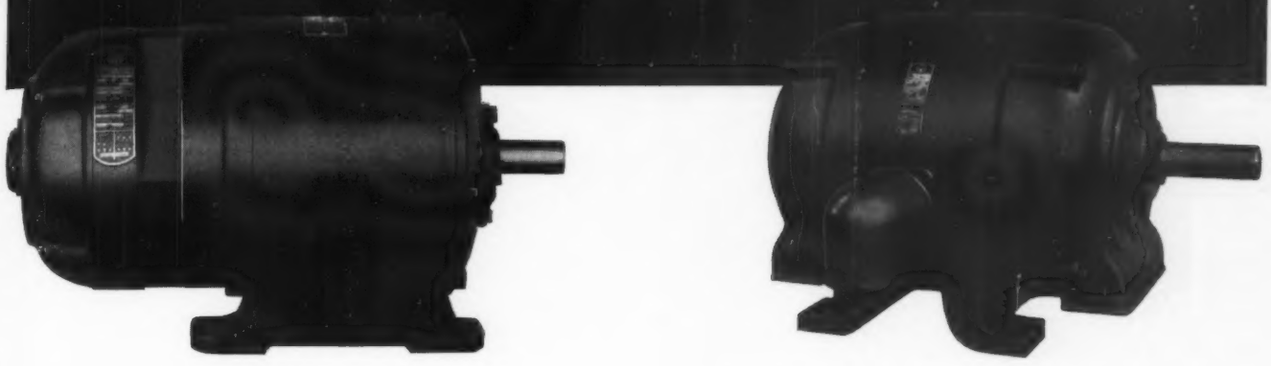


In addition, a built-in terminal box eliminates the necessity for splicing and soldering leads. Three binding posts are provided for direct connection to the line and cold-control leads of refrigerators. To protect the motor from falling dirt and water, drip-proof end shields have been used.

Gage Reads by Small Increments

DESIGNED for use where inspection, testing, etc., are held to a finer degree of accuracy than is possible with dial gages reading in thousandths, the new dial gage No. 726-A of Brown & Sharpe Mfg. Co., Providence, R. I., is graduated to 0.0001-inch. The gage, equipped with jewelled bearings, is regularly furnished with a lug on back with a 3/16-inch hole. The dial is 2-7/32 inches diameter and has a 0.020-inch spindle movement. The pointer of the gage is

AGAIN!—New F-M Motors give you a Sales Advantage!



GEAR MOTORS

from 2 revolutions per minute up

Where standard motor speeds are too high—where you need a compact, simple, slow speed drive—*there* is where you need a Fairbanks-Morse Gear Motor.

The F-M Gear Motor is the same high quality F-M Ball Bearing Motor with standard NEMA frame, built integrally with a 97% efficiency speed reducer which is also equipped with anti-friction bearings.

Either motor or gear assembly may be removed for inspection or repair *quickly*—and re-alignment is positive and accurate due to an indexed fit between the two units.

Cost is considerably less than the conventional motor, coupling and gear box assemblies.

Complete information mailed immediately on request.

SPLASH-PROOF MOTORS

Protected Against All Liquids and Falling Particles

Something new in practical design of splash-proof motors—and far superior to the usual "adaptation" to this kind of service. From its very beginning the F-M Splash-Proof Motor was designed to be what it is named—*splash-proof*—fully protected against steam, water or other liquids which may be directed toward it and against all falling particles.

The only openings are the flared air ducts which open at the bottom slightly above the level of the motor feet.

These air ducts are fitted with removable, perforated baffles which prevent the entrance of any water which may be directed into them.

The design of these baffles is such that resistance to air is negligible. Cooling air is equally and thoroughly distributed, resulting in a sweet running motor with no "hot spots."

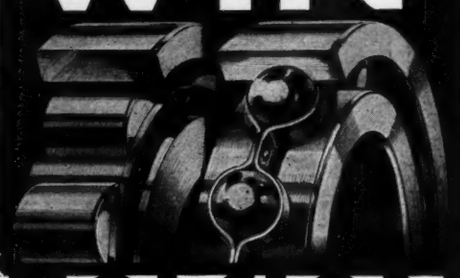
Want to read all about a truly Splash-Proof Motor? Send for descriptive bulletins just off the press. Fairbanks, Morse & Co., 900 S. Wabash Ave., Chicago, Illinois.

Pioneer Designers and Manufacturers of POWER, PUMPING AND WEIGHING EQUIPMENT 104 Years

FAIRBANKS-MORSE

MOTORS

5976-EA 40.76

TWIN

PRECISION

IN the past 20-odd years many standards have become flexible, and quality has—in many cases—become a variable thing . . . But throughout this period NORMA-HOFFMANN Precision Bearings have been consistently made to the highest standard of excellence . . . They continue to be the choice of those who measure value by service rendered, and who seek the lowest cost per bearing per year of useful life . . . *Write for the Catalog. Let our engineers work with you.* . . . . . .

"NORMA-HOFFMANN"
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BALL, ROLLER AND THRUST
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STAMFORD, CONN., U. S. A.

removable and pointer of different forms and lengths can be used. Dial face may be turned to bring zero under the index pointer.

Introduces Vertical Motoreducers

MOTORIZED speed reducers in vertical models have been added to the line of Falk Corp., Milwaukee. The IX type for double or triple reduction, shown herewith, is an integral unit corresponding to the I type horizontal motoreducer. One motor end bell is removed

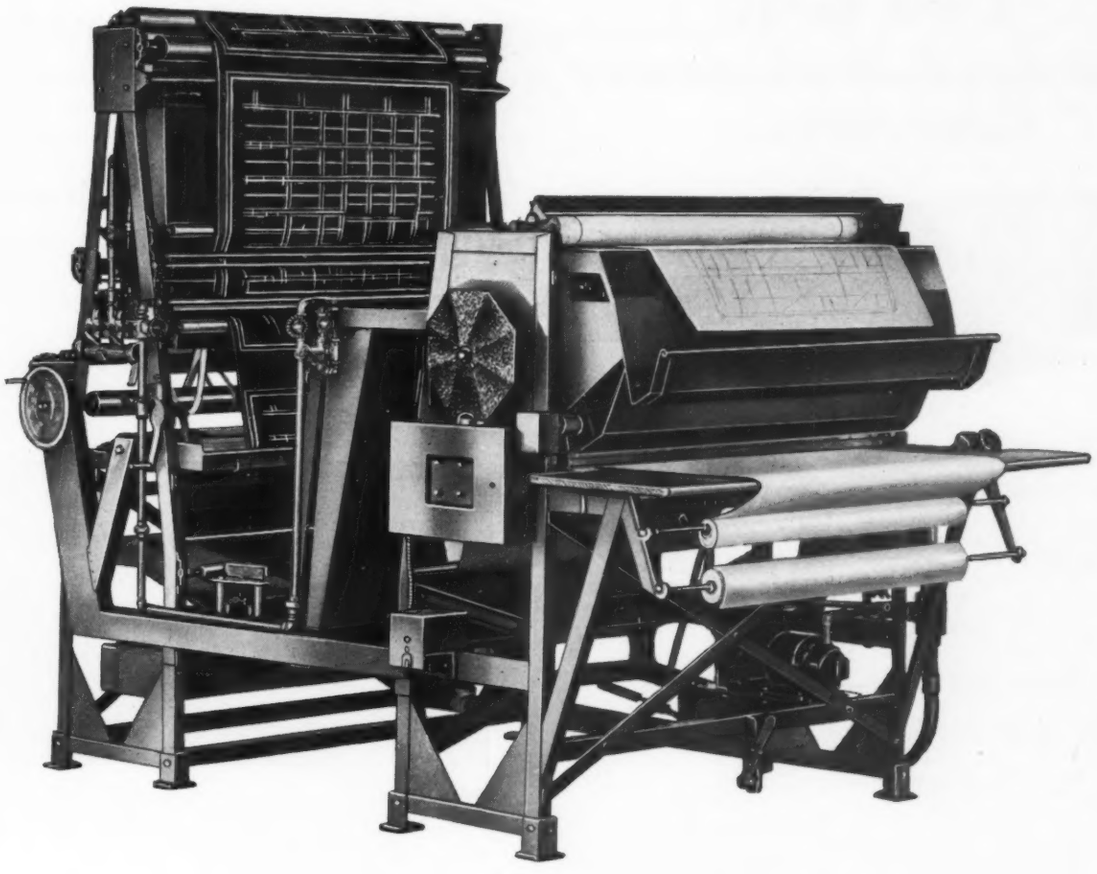
Stator is close coupled through an adaptor to the gear case in internal model of vertical motorized reducers

and the stator close coupled, through an adapter, to the gear case.

The ZX type is designed so that any standard horizontal ball bearing type motor with feet can be used unchanged, with resilient connection through a coupling to the gearing. Ratios are from 9.7 to 41.9 in double reduction and from 45 to 288 in triple reduction. The LX type is for single reduction only, 1.5 to 9.0, and corresponds to the geared-head L horizontal motoreducer.

Potentiometers Are Balanced

PHOTOELECTRICALLY balanced recording potentiometers in a radically new type have been introduced by C. J. Tagliabue Mfg. Co., Park and Nostrand avenues, Brooklyn, N. Y. In these instruments, a sensitive mirror-galvanometer is the primary controlling element in which an inertialess beam of light takes the place of the customary metal boom or pointer. The beam of light from the galvanometer to a photocell is interrupted by right and left shutters operated in synchronism with electric contacts in an electromagnetic circuit, through which in turn are operated the moving contact of the Wheatstone bridge or potentiometer, and the recording and controlling mechanisms. The photocell is not



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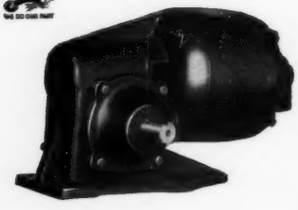

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070-8

a calibrated element but serves only to detect the direction of the light beam and bring the galvanometer to zero deflection, according to the null method of balancing an electric circuit. The new instrument, therefore, is not a "photoelectric potentiometer" in the sense that a balance of photoelectric currents is implied.

Stock Chains Are of Stainless Steel

STAINLESS steel chains as a regular production item in the most commonly used sizes are now being stocked by Diamond Chain & Mfg. Co., Indianapolis. These chains, shown herewith, are made entirely of 18-8 stainless steel throughout. They are recommended for use

Stainless steel chains resist corrosive action of fruit and vegetable juices

wherever corrosion is a problem either from the standpoint of service life, of sanitation, or of appearance. They resist the corrosive action of fruit and vegetable juices, do not discolor even in food-industry applications where the chain must pass through the processed food, and they are easy to keep sanitary.

Nickel Finish Has Great Brilliance

DEPOSITS of great brilliance and beauty are said to be obtainable with the new electrodeposited nickel finish developed by Metal Finishing division, Pyrene Mfg. Co., Newark, N. J. Subsequent polishing or color buffing is eliminated by the new finish. The finish can be applied in thick deposits and results in a surface about 10 per cent harder than the usual nickel deposit.

Introduces Inverted Converters

INVERTED converters, ranging in size from ½ to 20 KVA have been developed in a new line by General Electric Co., Schenectady, N. Y., for use with small machine tools, public address sys-

A successful gear steel
that is easy on cutting tools

ILLINOIS
S·A·E· 5140

This steel has actually doubled the life of cutting tools for some users. It is so free from hard non-metallic inclusions that it produces no peeling of the tool.

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SUBSIDIARY OF UNITED STATES STEEL CORPORATION
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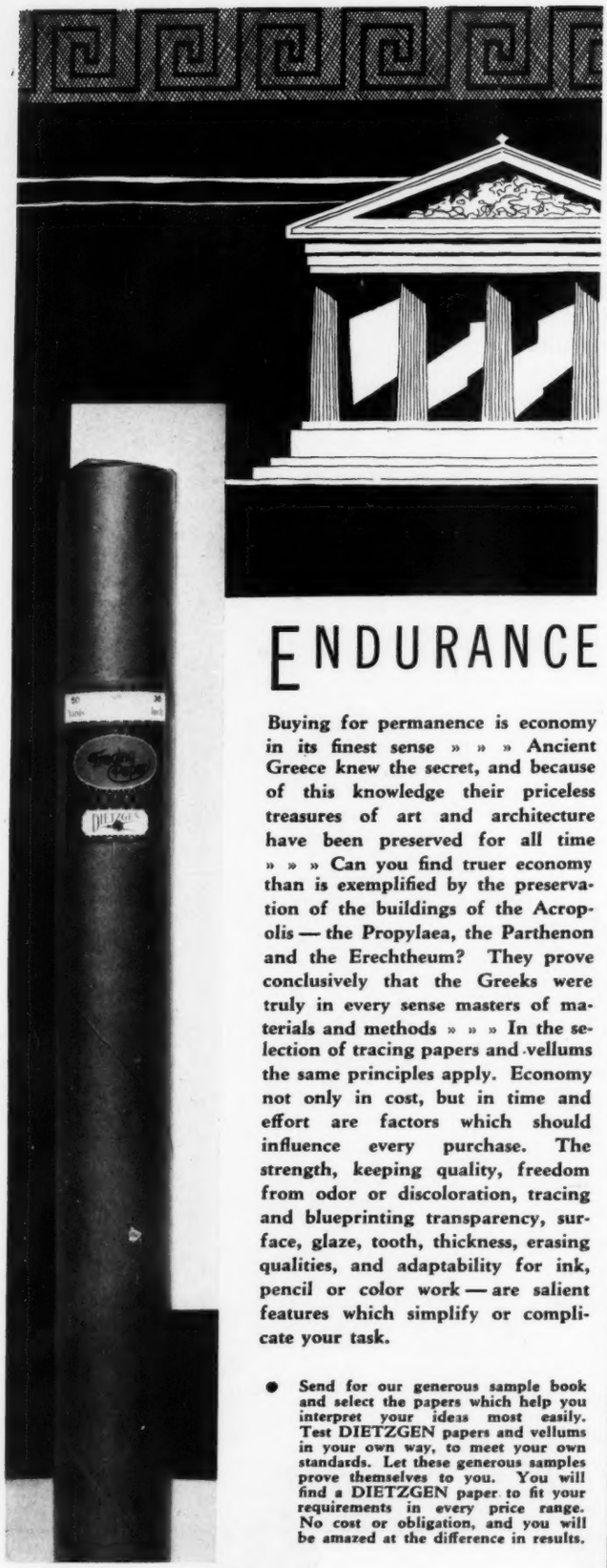

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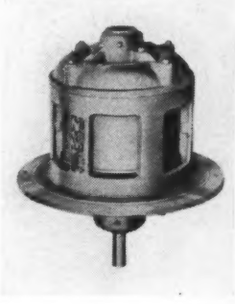
tems, and other applications where only direct current is available. The converters are available for changing 115 or 230 volts direct current to 110 or 220 volts alternating current. The field structure of this new equipment serves both windings and is identical with the field construction of standard direct current motors and generators. These two-bearing units use duplicate end shields. More compact than a standard motor-generator set, the inverted converters are readily portable.

Develops Flange Head Mountings

SEVERAL types of flange head mountings have been developed by Louis Allis Co., Milwaukee, for its line of electric motors. The flange shown in the accompanying illustration is an integral part of the motor head and may be finished on either or both sides. Limitations in space for motor mounting on many applications such as machine tools, including lathes, drills, grinders, polishers, etc., make it desirable to eliminate motor feet and provide finished flanges on the motor head for end mounting. This new type of motor is also used on close-coupled pump mountings.



Finished flanges on the motor head are provided for end mounting when space is limited

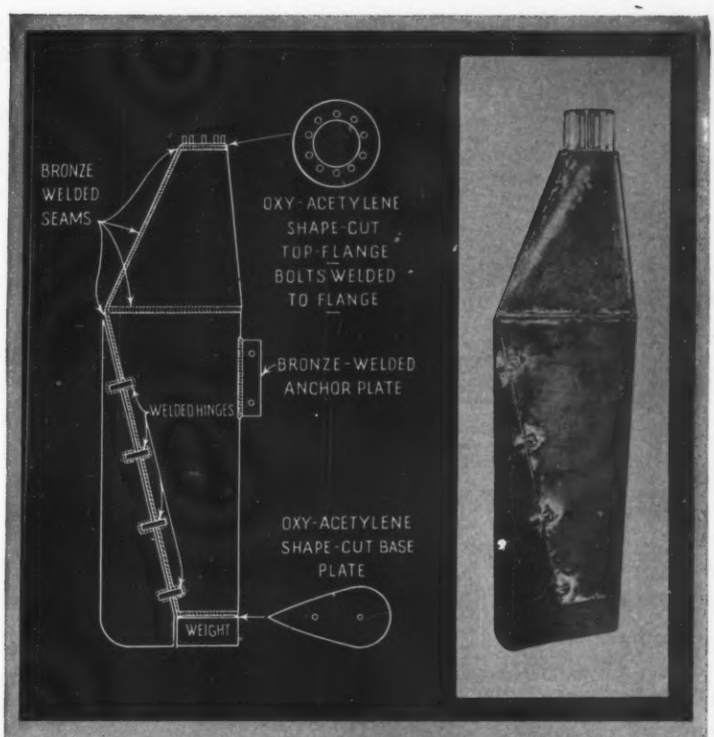
Mercury Switches Are Employed

MERCURY switches are employed to eliminate open make-and-break contacts in the new automatic control indicating thermometer recently announced by Brown Instrument Co., Philadelphia. In this controller, shown herewith, every six seconds a motor-driven control table determines the location of the pointer in reference to the control setting and tilts the mercury switch from one side to the other if the temperature has changed.

With this system the measuring mechanism is free to position itself, unhampered by the control mechanism. There is no excess weight on the pointer, no distortion or restriction of pointer movement. The making of control contacts is not dependent upon friction, because the

for Production Speed and Low Cost

Redesign for OXWELDING



Through the cooperation of Linde Process Service, the manufacturer of these river buoys was able to organize his shop for production oxwelding and to make a profit on the first lot. The suggestion of the Linde Service Operator that bronze-welding be used resulted in a strong, rustproof joint, greater speed in manufacture, and an estimated saving of over 50 per cent for gases and labor.

PRODUCTION costs of these river buoys were not low enough to meet contract prices and still produce a profit. This manufacturer, seeing the need for better metal-joining methods, called on Linde Process Service to help redesign the buoys for oxwelding.

On Linde's recommendation the job was laid out for oxwelding with bronze. Linde Service Operators helped to train and organize the welders to do the work under Procedure Control. Jigs were designed and built to insure fast, accurate production. The result—the contract was completed, the buoys delivered ahead of the specified time, and the manufacturer made a satisfactory profit. A repeat order for additional units confirmed the customer's approval of the job.

In this typical incident Linde Process Service employed only one of the hundreds of the oxy-acetylene welding and cutting application principles with which many years of experience in the development of the oxy-acetylene process have made it familiar. If you are designing or redesigning, the trained Linde representative, who will call at your request, may be able to suggest money-saving ideas which can be used in your manufacturing operations.

A letter or telephone call to the nearest Linde Sales Office will assure you the cooperation of this service. There is no obligation.

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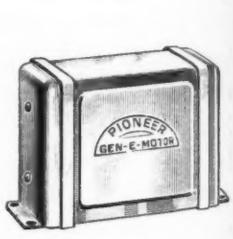
switch is positioned with a positive action by the motor. Contacts are unaffected by vibration as the mercury switch is mechanically locked in place until a different position is required. The

Measuring mechanism is free to position itself unhampered by control mechanism in improved indicating thermometers

mercury switches have capacities up to 15 amperes at 110 volts, 10 amperes at 220 volts, thus eliminating external relays in most applications.

New Small Converters Are Announced

CONVERTERS for operating any small alternating current appliance from direct current sockets have been developed in new compact designs by Pioneer Gen-E-Motor Corp., 464 West Superior street, Chicago. The unit shown herewith measures only 4⅛ inches high by 2⅞



Small alternating current appliances may be operated by direct current through new converters

inches deep and is furnished complete with cord and plug for connecting to any 110 volt direct current socket and will then provide a 110 volt, 60 cycle alternating current output. Models are also available for operation from 6 and 32 volt direct current sources.

Design Tables Conserve Space

CONSERVATION of space and maximum convenience is assured by the auto-shift drawing and reference tables developed by Hamilton Mfg. Co., Two Rivers, Wis. The tables, shown herewith, are designed to be arranged in series, so that each designer has use of the adjustable

YOUR HOURS OF RECREATION OWE MUCH OF THEIR PLEASURE TO NICKEL

WORK gives plenty of jobs to Nickel. But so does PLAY.

Consider some of your favorite diversions. Through the aid of Nickel many of them are made far more pleasant and enjoyable.

For example take your radio. Every metal part of its screen grid tube is made of Nickel or contains Nickel as the principal ingredient. And the tone range would not be so broad nor the reception so sharp and clear were it not for Nickel. The laminated plates of the cores of the better radio transformers are made of an alloy high in Nickel.

Those color pictures you enjoy so much at the movies. They also owe much to alloys containing Nickel. For Nickel-bearing metal provides equipment that is rust-free, corrosion-free and highly resistant to chemical attacks... properties that are indispensable in the apparatus used for developing and transferring color films.

When you ride in your car, Nickel again comes to your aid. Strong, tough, hard alloys of Nickel contribute in various ways to its speed, safety, dependability and long life. Still other alloys of Nickel as well as heavy Nickel plate give trim and hardware their bright, durable, silvery lustre.

Aboard your motor boat Nickel irons and Nickel steels minimize the risk of engine failures, while gleaming Nickel silver and other Nickel-alloyed decorative metals provide corrosion-free bright-work. Galley equipment is made of silvery Monel Metal, a high Nickel alloy, while underwater a propeller shaft of this same sturdy corrosion resistant alloy smoothly transmits the power of the engine to the Monel Metal propeller.

Nickel...

IN YOUR RADIO

Every metal part of the screen grid tube is either made of Nickel or contains Nickel as the principal ingredient.

Even when you wade your favorite trout stream, Nickel helps you cast your fly. For Nickel is widely used in reels, rod ferrules and trimming. It is also used in braided trolling lines, lures, etc.

Not only in the field of Recreation, but in every field where man uses metals, there is an alloy containing Nickel that can be employed to fill some particular role more dependably and more economically.

These alloys embrace a variety of compositions, each developed and firmly established as a result of laboratory and service tests. They offer a wide range of improved properties... properties that assure increased resistance to heat, stress, fatigue, erosion, corrosion and wear.

Among them are Nickel Alloy Steels, Nickel Cast Irons, Nickel Bronzes, Stainless Steels, etc... all of which are commercially available in every important metal consuming center. If you are planning to build new equipment or modernize the old, our engineers will be glad to advise you concerning the most suitable applications.

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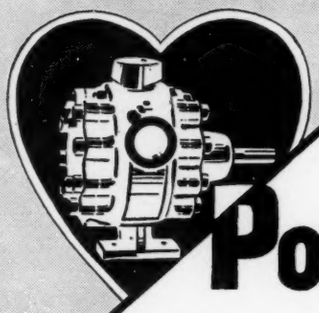

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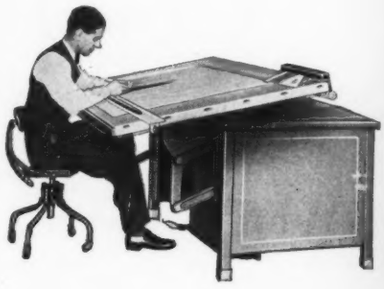
BRUNING

drawing board in front of him, with a reference-desk surface including tool drawer and reference drawers behind him.

The drawing table is held rigidly at any position or any height by two systems of brakes,



A system of brakes holds the drawing board rigidly in any position or at any height

which work independently. A small foot pedal releases the brakes, and then the board can be raised or lowered easily. To change the slant of the board the tilting mechanism brakes are released by lifting a rod under the front edge of the board, and the board can be tilted to any desired position. The board is balanced dynamically so that all adjustments can be made with a minimum of effort.

Ceramic Materials Solve Corrosion Problems

(*Concluded from Page* 25)

ceramic materials into one leak-proof rotating element. *Fig.* 3 shows a section through the gland. The gland shroud sleeve must be made of a wear resistant ceramic material capable of withstanding high mechanical loadings and severe thermal shocks. The material used for the sleeve is Sillimanite. It is extremely hard to machine by grinding and is unsuitable for use in the impeller. Its use as an ideal shroud makes a joint on the rotating shaft with the impeller necessary. The shroud which is ground to close tolerances all over is joined to the impeller with a hard gasket joint. It is free to move longitudinally on the metal shaft but is held against radial slippage by a key which passes through it. A metal collar that backs the sleeve at the power end is also keyed to the shaft but is free to slip longitudinally. The collar is separated from the Sillimanite sleeve by a hard gasket.

Torque Action Tighten Threads

When the impeller, sleeve, collar and stub shaft are screwed into the alignment counterbore and internal thread of the main pump shaft, the whole assembly is pulled together and all ceramic parts and gaskets kept in continual compression. Internal threads automatically take up any continued compression of the

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WHEN high pressure rods are to be packed so they will stay packed, engineers specify one of the several types of Garlock Shredded Metal Packings.

Garlock 7050 and Garlock 7050-C are two of the most prominent of these shredded metal packings. Both types are giving remarkably long service on air and gas compressors—in many instances against pressures up to 3,000 pounds, or higher. These packings are being used with equal success against hot water at high pressures and on other conditions where temperatures are not in excess of 625°F.

The shredded metal rings in Garlock 7050 and 7050-C are alternated with soft metal separator rings. Specially constructed top and bottom adapter rings complete the set. For CO_2 service bronze end rings usually are furnished.

A Garlock representative will gladly tell you more about these Quality Controlled Garlock Products.

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Hardened and Ground Ball Thrust Bearings carried in stock in a wide range of sizes for immediate delivery

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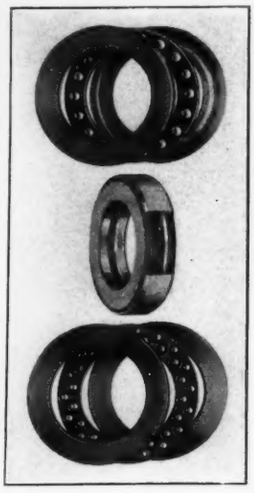

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36th Year

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NO POWER TAKE-OFF NEEDED

This Pump is self-contained

These compact, geared motorpumps fit into many a "tight" place—no power take-off is needed. They simplify and make possible improvement in machine design.

Made in 3 sizes for Vertical and Horizontal Mountings. Circular on request.

Brown & Sharpe Mfg. Co., Providence, R. I., U. S. A.

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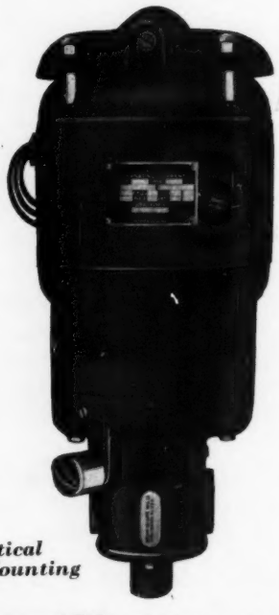

Vertical Mounting

Geared Motorpumps
Nos. 101, 102 and 103

Brown & Sharpe Pumps
Geared — Vane — Centrifugal

gaskets at the ends of the shroud sleeve as these threads are constantly tightened by the torque action of the drive. The one disadvantage of this arrangement lies in the fact that if the pump were driven in the wrong direction the impeller assembly would unscrew and the impeller would be driven against the case cover. This is prevented by a reverse phase relay incorporated with the driving motor which prevents reversal of the drive.

In other stoneware pump gland designs much trouble was experienced with the stoneware gland followers binding due to improper adjustment of gland pressures by attendents in charge of maintaining pumps. These designs allow acid seeping from the gland to spray out and cause damage to the exposed metal parts as well as endanger people standing close by. To overcome this a dual gland was provided with spring tension members which keep the gland pressure approximately constant in spite of temperature fluctuations. The rear or auxiliary gland acts as a wiper and directs all seepage into a small stoneware drain cup. This arrangement is most useful when new packing is being run in, at which time a reasonable amount of seepage is expected.

Cast Iron Shell Used

To prevent binding of the chemical stoneware gland followers a cast iron outer gland shell is used. This slides longitudinally on a machined extension of the front bearing cap. The inside surfaces of the gland shell are machined to form a sliding fit wifh the outer surfaces of chemical ware glands. The whole assembly is adjusted by changing the compression of two coil springs which tend to force the gland assembly toward the packing. Four coil springs located in chambers of the iron gland shell maintain a constant pressure on the auxiliary gland follower. It is obvious that binding of the gland by excessive tightening of one adjusting nut cannot pull the assembly out of line as it is free to move in a longitudinal direction only, due to the guide formed by the bearing cap extension. This arrangement has proved most satisfactory in actual use and has permitted efficient use of chemical stoneware to accomplish a difficult gland job.

When size of equipment prevents the use of chemical ware for all the parts of a machine which come in contact with corrosive elements, it is sometimes possible to use both stoneware and stainless steel. This has been done in the case of large fans where the tensile stress in the runners would be too high for stoneware. Under such a circumstance the case can be made of stoneware and the runner of stainless steel. Such a combination reduces to a minimum parts that must eventually be replaced due to corrosion.

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HEAVY LOADS·· at HIGH SPEEDS

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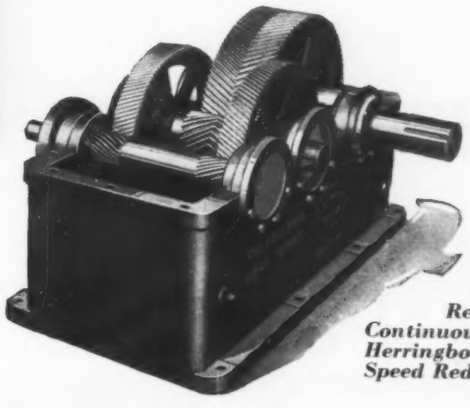
Philadelphia

Continuous Tooth

Herringbone Gear Speed Reducing Units



Double Reduction Continuous Tooth Herringbone Gear Speed Reducer

These Speed Reducing Units, using gears of the Sykes tooth form, have greater gear strength and extra load carrying capacity. The units combine silence, strength, long life and low maintenance cost.

Write for complete details.

Philadelphia MotoReduceR (Horizontal Type)

and don't forget — —

The Philadelphia MotoReduceR is a most efficient and compact means of driving plant equipment. It combines a motor and speed reducing unit in one complete housing. The many construction and operating features are described in the MotoReduceR Catalog. Send for a copy.

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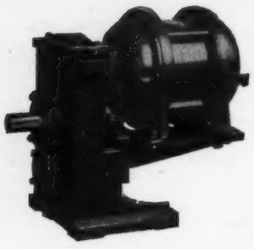


because of the flexible chain and sprocket drive

The Cullman Speed Reducer takes but little more space than the motor. Its compactness is due to the chain and sprocket drive which effects the first speed reduction and cuts the first gear contact speed from two to six times the motor speed.

The chain and sprocket drive is flexible and operates with a minimum amount of noise. The chain also serves to keep the gearing and bearings perfectly lubricated.

Direct connected . . . no exposed moving parts . . . oil and dust proof.

Cullman Speed reducers from 1/8 H.P. to 16 H.P. Write for booklet containing complete information.

Cullman Wheel Co.
1356 Altgeld St., Chicago, Ill.
CULLMAN SPEED REDUCERS

adaptable to scores of applications

Viking ROTARY PUMP

THE three general classes of Viking Rotary Pumps will meet most any requirements you may have. All three employ the Original Viking "Two Moving Parts" Principle. The Standard Pumps in capacities of 1/5 to 1,050 G. P. M. are ideally adapted to the economical handling of all grit-free liquids. Viking Hydraulic Oil Pressure Pumps offer low power requirements and wide adaptability in the operation of hydraulic lifts and elevators, actuation of machine tool movements, etc. Made of laboratory-tested metals in capacities of 5 to 45 G. P. M. Viking Coolant Pumps in capacities of 5 to 35 G. P. M. are directly reversible. Write for FREE Folder and Prices.

Viking Pump Company
Cedar Falls, Iowa

COMPOSITION MATERIALS—General Plastics Inc., North Tonawanda, N. Y., has written a bulletin on molding compounds especially for designers and engineers. Advantages of the compounds and their limitations in design are given together with drawings showing the right and wrong procedures. Also included is a description of the materials and their properties, and possible applications.

CONTROLS (ELECTRICAL)—Struthers Dunn Inc., Philadelphia, is distributing a folder on its line of relays which have contacts of silver. A description and illustrations of the relays are included.

CONTROLS (ELECTRICAL)—Switch and Panel division of Square D Co., Detroit, is distributing new catalog pages for its loose-leaf catalog No. 37. The new pages discuss switches, fuse cabinet and switch combinations, circuit breakers, panels, meter boxes and other equipment.

COUPLINGS—A new general purpose coupling, known as type WH flexible coupling, is presented in a catalog insert of Westinghous Electric & Mfg. Co., East Pittsburgh, Pa. The coupling is designed for use with gearmotors, speed reducers, and other industrial drives. It requires no lubrication. Ratings and dimensions of the couplings are included in the publication.

DESIGN DEPARTMENT—Blueprint machines that can be used to reproduce blueprints, Van Dyke negatives, blue line prints, black and white prints or Ozalid prints are announced in a bulletin of Shaw Blue Print Machine Co., Newark, N. J.

DRIVES—Sterling Electric Motors Inc., Los Angeles, has prepared bulletin No. 133 on its Vari-speed motors in upright enclosed design.

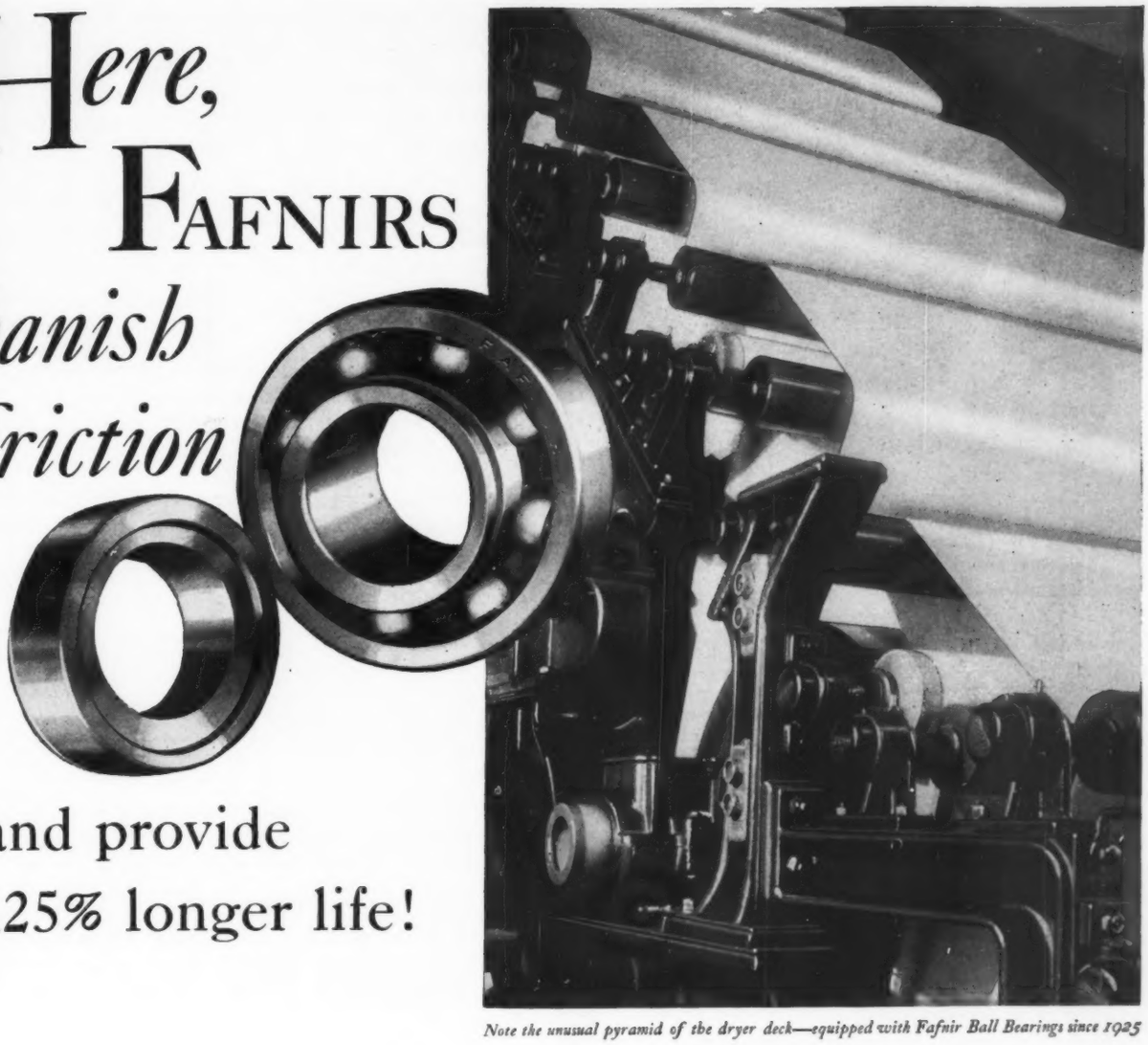
DRIVES—Vertical motoreducers in several types are presented in bulletin 271 of Falk Corp., Milwaukee. The new units are described and pictured, and advantages of their application are given.

GAGES—Brown & Sharpe Mfg. Co., Providence, R. I., is distributing a circular describing its new gage which is graduated to 0.0001-inch.

GEARS—A data book of engineering and gear information has been published as catalog No. 57 by Charles Bond Co., Philadelphia. Thousands of gear sizes in a number of different materials are listed. Included also is a new line of speed reducers made in sizes from fractional to 5 horsepower, and data on sprockets.

GENERATORS—Direct current generators and exciters are described in a catalog insert of General Electric Co., Schenectady, N. Y., which includes photographs of

Here, FAFNIRS *banish friction*



and provide 125% longer life!

Note the unusual pyramid of the dryer deck—equipped with Fafnir Ball Bearings since 1925

Back in 1925 the Keith Paper Company of Turner's Falls, Mass., installed Fafnir Ball Bearing Pillow Blocks on a new dryer. Paper and felt-carrying rolls were mounted on these friction-free bearings. Result—so much less power needed to turn the rolls by the web of paper that web breakage was greatly reduced. Moreover, the life of the felt which was formerly four months is now *nine months.* Now Keith tell us that these bearings "have proven 100% successful . . . "

Power is obtained from a 30 H.P. motor driving six variable speed gear units. Twelve Fafnir Ball Bearings support the shaft extensions. Six more carry the pinion shaft. And in the five sectional driving units more Fafnirs are used. To quote Keith again, "In the eight years of continuous service not one of these (Fafnir Ball Bearings) has been disassembled for repairs or maintenance."

Not only do Fafnirs save power and lubrication. They minimize maintenance and promote dependable, profitable operation. One of the most certain steps towards profitable manufacture is the elimination of friction and waste. Give Fafnir anti-friction bearings the job of keeping shafts, wheels and gears running freely. The work of replacing plain bearings is simple—the profit worthwhile. THE FAFNIR BEARING COMPANY, *New Britain, Conn.*
Atlanta . . Chicago . . Cleveland . . Dallas . . Detroit
Milwaukee . . Minneapolis . . New York . . Philadelphia

SAVINGS *through successful fighting of friction, in machines of all kinds on land and sea and in the air, are recorded in* "THE DRAGON". *We will gladly put your name on its mailing list. There's no obligation.*

FAFNIR BALL BEARINGS

LAMINUM SHIMS in slotted bushings provide .002 in. accurate adjustment. No miking. No filing. Just *peel* the laminated shim.

Minimum maintenance and longer life are assured in this Mac-Roh bread slicer. The slotted bushing shown is one of 5 Laminum-equipped bearings in this machine.

LAMINUM

LAMINATED SHIM COMPANY, INC.
2126 Forty-fourth Ave., Long Island City, N. Y.

517

YOU'LL WANT THIS NEW GEAR CATALOG

IT'S FREE • SEND FOR IT!
Thousands of sizes of STOCK Gears. Also new Line of Speed Reducers.

CHARLES BOND COMPANY
617 A Arch Street Philadelphia, Pa.

When requesting this Catalog, please mention MACHINE DESIGN

the parts used in the generators and an outline of their features.

INSTRUMENTS—Indicating thermometer controllers which employ switches to eliminate open make-and-break contacts controlled by the pointer, are presented in catalog No. 6702 of Brown Instrument Co., Philadelphia.

LUBRICATION AND LUBRICATING EQUIPMENT—Colloidal-graphited lubricants, what they are, how they benefit the machine, how they may be applied most successfully and similar information is given in bulletin No. 92.6 of Acheson Colloids Corp., Port Huron, Mich.

MOTORS—Allis-Chalmers Mfg. Co., Milwaukee, has issued a bulletin on its line of squirrel cage induction motors. Features of the motors are presented and individual parts are shown by photographs.

MOTORS—Direct current motors are the subject of two new loose-leaf bulletin sheets, known as bulletin 167, part 8, recently prepared by Wagner Electric Corp., St. Louis. The bulletins are profusely illustrated and all motor design and construction details are fully described.

MOTORS—Positive protection against burnouts is built into the windings of the self-protecting motors described in a recent bulletin prepared by Lincoln Electric Co., Cleveland. The bulletin relates how this protection is obtained and includes ratings and dimensions in all frame sizes.

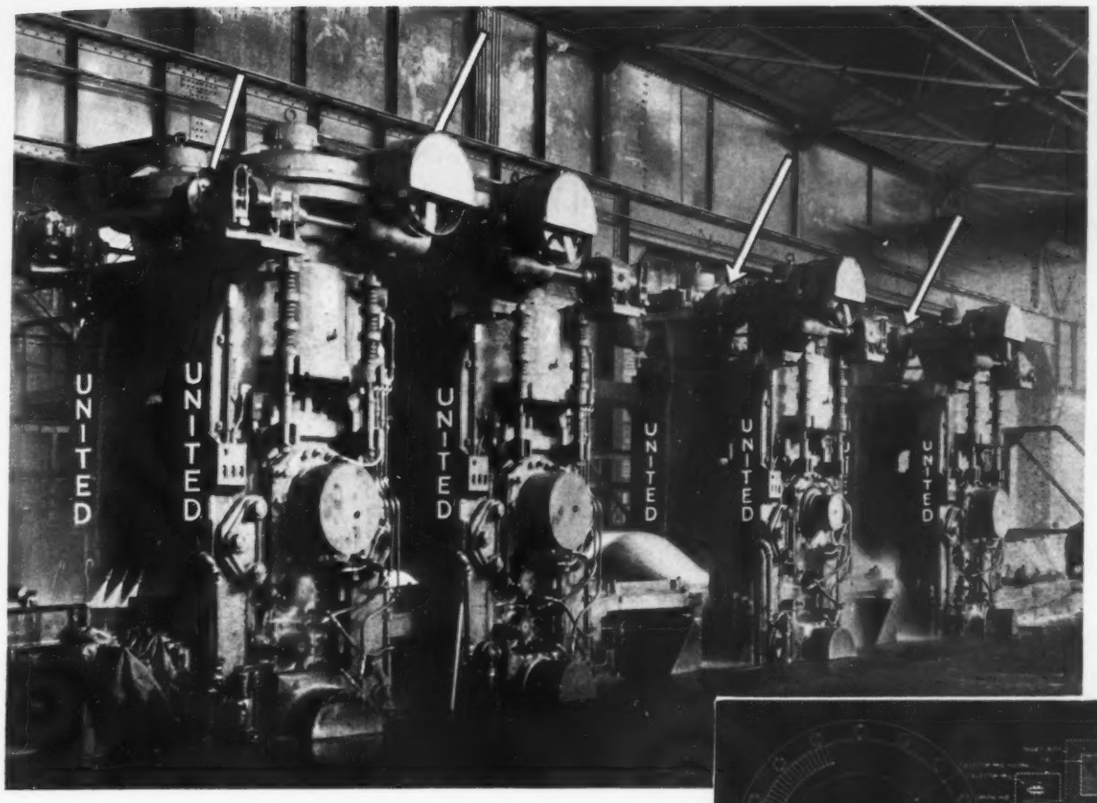
PUMPS—Sizes and displacements of internal bearing, double helical rotary pumps are given in a data sheet issued by Worthington Pump & Machinery Corp., Harrison, N. J.

SHAPES—The determination of the choice of steel, especially as related to cold drawn shapes, is discussed in a recent bulletin of Union Drawn Steel Co., Massillon, O.

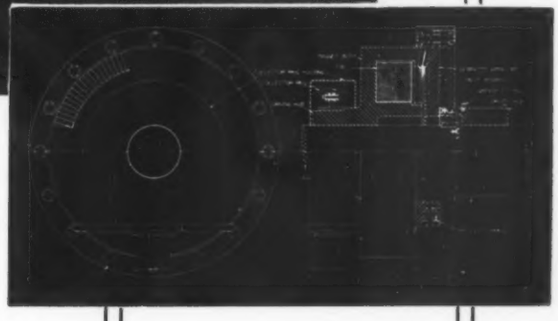
TUBE COUPLINGS—Complete information on tube couplings and associated equipment of many types is given in bulletin No. 35 of Parker Appliance Co., Cleveland. The parts are illustrated and described and applications are presented in detail.

Research Publications

Some Physical Properties of Platinum-Rhodium Alloys. In a continuation of the investigation of metals of the platinum group, the bureau of standards has completed a study of platinum-rhodium alloys of high purity, ranging in composition from pure platinum to pure rhodium. The melting point, hardness, density, electrical resistivity, temperature coefficient of resistance, and thermal electromotive force against platinum were determined for each alloy. The microstructure of each alloy appeared to be that of a solid solution. The high melting points, electrical properties and resistance to oxidation of the platinum-rhodium alloys should lead to their use, particularly the alloys containing 20 to 40 per cent of rhodium, for furnace resistors for service at high temperatures. Published as RP 650. Available through superintendent of documents, government printing office, Washington.



Dings serrated clutch couplings installed on the screwdowns on a 4-high finishing mill.



Dings Magnetic
CLUTCHES SOLVED THIS PROBLEM

What about Yours?

Dings magnetic clutches on the screwdowns of this finishing mill solved the problem of absolute synchronization of the drive units. When separate motors are used, the worm shafts can be positively tied together and creeping is eliminated.

What is your clutch problem? Perhaps it's one of remote control! Perhaps it's a question of saving space! Perhaps it's a matter of economy!

Magnetic clutches have outstanding valuable advantages. They assure *instant* engagement, they stop with overload, they can be used with high speed drives, they are simplicity itself, save space and protect the operator.

Before figuring on your next clutch installation, talk to one of our engineers. There is one near you and his help entails no obligation.

DINGS MAGNETIC SEPARATOR CO.
666 Smith Street, Milwaukee, Wisc.

The Experience of the Oldest Manufacturer of Magnetic Equipment can be made worth while to you . . . .

The Dings Magnetic Separator Co. is the oldest manufacturer of magnetic equipment—since 1899 Dings Engineers have been developing magnetic appliances for all kinds of transmission, separation and special problems. Here is the greatest magnetic experience that can be secured in promptly solving your clutch problem. There is no cost in taking advantage of it.

Dings *High Intensity* MAGNETIC CLUTCHES

EXPERIENCE THAT GOES BACK TO 1899

Dumore Tools will be exhibited in booths 7, 8, and 9 in Group T, Building No. 3 of the General Exhibits Group.

Dumore motors are made in sizes up to ¾ H. P. In the smaller sizes they are available with built-in gear reduction units — gear ratios from 1089:1 to 2⅝:1 and gear shaft speeds from 4 to 3,000 R. P. M. Write for catalog giving complete data on sixty different sizes and types of Dumore Universal Motors.

THE DUMORE COMPANY, 100 Sixteenth St., Racine, Wis.

DUMORE
FRACTIONAL HORSEPOWER
MOTORS

. . . . . In the MARLIN-ROCKWELL CORPORATION'S Advertisement which appeared in the May issue, page 45, two typograpical errors were reproduced.

The fifth line of the text matter, reading "Strom . . . . . with 20 years", should read "Strom . . . . . with 25 years." In the line "What 89 Years of Specialization" appearing in the head, the figure 89 should read 36.

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• • •

MACHINE DESIGN is a monthly technical publication conceived, edited and directed expressly for those executives and engineers responsible for the creation and improvement of machines built for sale, and for the selection of the materials and parts to be used.

Business and Sales Briefs

W H. HENRY, manager of sales of the fractional horsepower motor section of the motor division of the industrial department, General Electric Co., is now located at the general office in Schenectady. He will assist C. F. Pittman in the administrative work of the motor division. A. W. Bartling has been appointed assistant manager of sales in this department with headquarters at Fort Wayne, Ind. P. O. Noble has been appointed engineer, and A. F. Welch consulting engineer of the department at Fort Wayne.

* * *

Chain Belt Co., Milwaukee, has removed its Detroit office to 5169 Martin avenue. The office will continue to be in charge of C. E. Martin. The Detroit warehouse of the company is now stocked with many types of chains and sprockets.

* * *

O. Kuhler, industrial designer, has removed his offices to 136 Liberty street, New York.

* * *

Charles Bruning Co. Inc. has opened a new sales office at 757 North Broadway, Milwaukee. Warren H. Coate is in charge.

* * *

The name of Acheson Oildag Co., Port Huron, Mich., has been changed to Acheson Colloids Corp. The organization, policies and location of the company remain the same.

* * *

Universal Power Equipment Corp., Cleveland, has appointed Welding Equipment & Supply Co., 3524 East Seventy-fourth street, Cleveland, its sales representative for northern Ohio for arc welders, electrodes and accessories.

* * *

Seth H. Taylor Jr. is now Pacific Coast manager for Lincoln Electric Co., Cleveland. For the last four years Mr. Taylor has been in charge of the Los Angeles office of the company. He will make his headquarters in San Francisco.

* * *

Edgar N. Dollin, formerly president of the Allied Die-Casting and Acme Die-Castings Corporations and formerly general manager of Precision Die Casting Co., has been appointed manager of the American Type Founders Corp., die casting division. This die casting division will be expanded so as to take in all forms of commercial die casting.

* * *

P. R. Dudley, formerly of Philadelphia, and closely identified with the metal working industry for the past five years, has been placed in charge of special sales work for the metal stamping division of Dickey-Grabler Co., Cleveland. Mr. Dudley's territory comprises northern Ohio and western Pennsylvania, southwestern New York and southern Michigan.

* * *

William J. Carroll has been appointed general sales manager of Gamewell Co., Newton, Mass., and its subsidiaries, which include Holtzer-Cabot Electric Co., Rockwood Sprinkler Co., and Eagle Signal Corp. Mr. Carroll has been vice president and general sales manager of the Rockwood division of the Gamewell company at Worcester, Mass., a position he leaves to take up his new duties.

* * *

Foote Bros. Gear & Machine Co., Chicago, has recently appointed new territorial representatives as follows: F. A. Emmons Jr., to one of the five divisions of the Chicago territories; C. C. Hanson, New Buffalo, Mich., as representative for the northern half of Indiana with the exception of Lake county; J. H. Nicholson, 6315 North Thirty-third street, Omaha, Nebr., as sales representative for Nebraska and the two western lines of counties in Iowa; John M. Davies, U. S. National Bank building, Denver, as representative in Colorado and Wyoming; S. D. Calloway, 3029 Roanoke road, Kansas City, Mo., as representative in the state of Kansas; and R. K. Plummer, 4015 Navahoe road, Cleveland, as direct factory representative in the northern half of Ohio.

Gear Experts Discuss Analysis

(*Concluded from Page* 62)

be used to prevent excessive local wear and sometimes local tooth breakage. This is a cautionary note. The answer to this problem is yet to be worked out.

—Earle Buckingham,
Massachusetts Institute of Technology

Expresses Essential Considerations

THE manufacture of helical and double helical gears for many years has been controlled by a few companies familiar with the operating conditions of the gears. The selection of sizes is largely dependent upon a certain intuition acquired by long familiarity of the subject and many attempts have been made to express this knowledge in terms of formulas but none have been entirely successful.

It is the writer's opinion that Mr. Schmitter's paper is an extremely valuable addition to the art since it expresses the essential considerations in formulas and is strengthened by simple explanation.

The enterprise of the Falk Corp. in promoting the necessary experiments and practical tests is to be highly commended and the generous manner in which this information has been given to the engineering world is significant of prominence in their selected field.

The subject is, of course, too intricate to be covered entirely by one paper, e.g., all of the considerations in Mr. Schmitter's paper are based upon gears produced with theoretical accuracy and it is obvious to even a casual investigator that the major problem confronting the manufacturer of helical and double helical gears is that of producing in metal gear teeth with accuracy of contour, division and helical trace with equivalent accuracy. This branch of the subject has absorbed the attention of all true gear specialists and has cost many hundreds of thousands of dollars. The writer is hoping, therefore, that Mr. Schmitter's most excellent paper be supported by a paper from a competent authority on gear machine design which will simplify the problems of the average engineer who obviously does not have time to investigate the many intricacies of gear design.

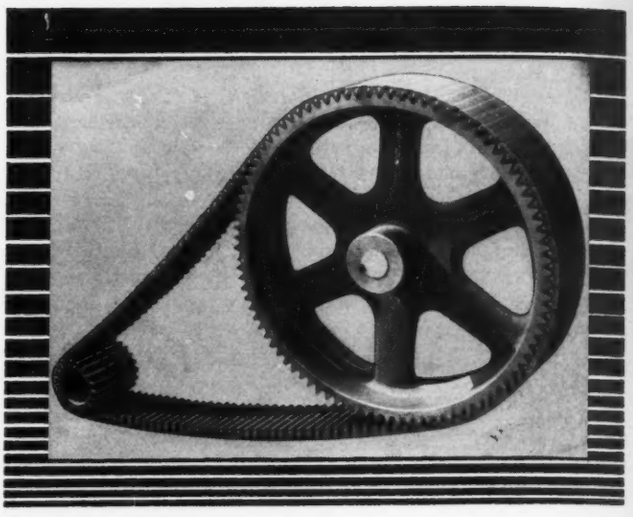
In conclusion, in the writer's opinion, the selection and use of special materials will play the most important part in gear design in the next few years.

—T. Holloway,
United Engineering & Foundry Co.

Additional comments by experts in the gearing industry will be presented with the second half of this article in the July issue of Machine Design.

FOR PERFECT FLEXIBILITY WITHOUT POWER LOSS USE

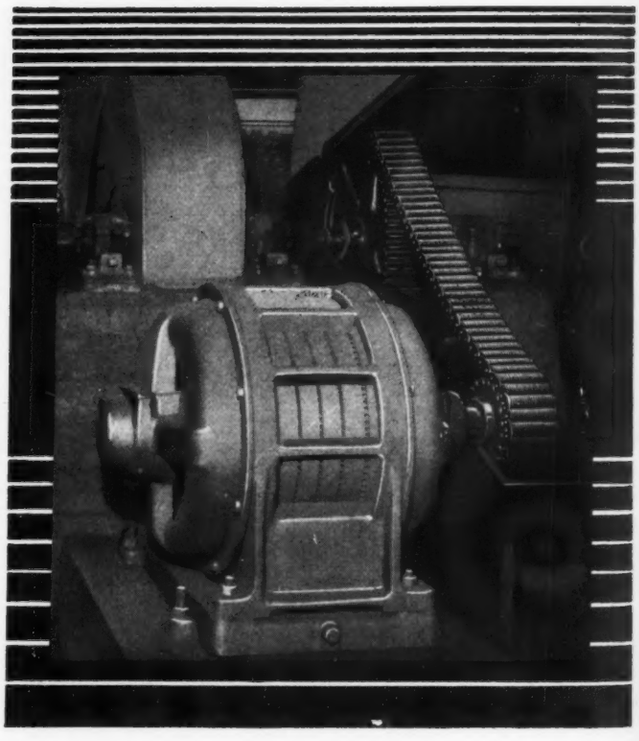
CHAIN DRIVES



Chain drives meet any transmission condition, can be applied to the centers you want; in fact, there's no form of transmission quite as satisfactory under all conditions.

For small drives or large ones, chain drives are superior over any other form of drive. They perform and, due to their low first cost and low maintenance cost, chain drives save money, decrease operating costs.

Our engineers will gladly cooperate with you in accomplishing definite results.



Silent Chain Drives
Morflex Couplings
Pullmore Clutches

MORSE CHAIN COMPANY

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Roller Chain Drives
Morse Kelpo
Free-Wheeling Clutch